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ge 369—Journal Section, American Society of Heating and Ventilating Engineers

Control of welding fumes is essential in shipyards and factories engaged in production for victory

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Announcing A New Uniform for the famous Red Belt

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Frank P. Keeney

Published monthly by the Keeney Publishing Co., 6 N. Michigan Ave., Chicago. Telephone: State 6916/ Subscriptions: U. S. and possessions, Canada, Mexico, Central America, South America, $2.00 per year; other, $4.00. Current copies, U. S. and possessions, 25c each, back copies 50c each. January, 1942, issue, $1.00 per copy/ Member of Audit Bureau of Circulations and the Associated Business Papers, Inc./

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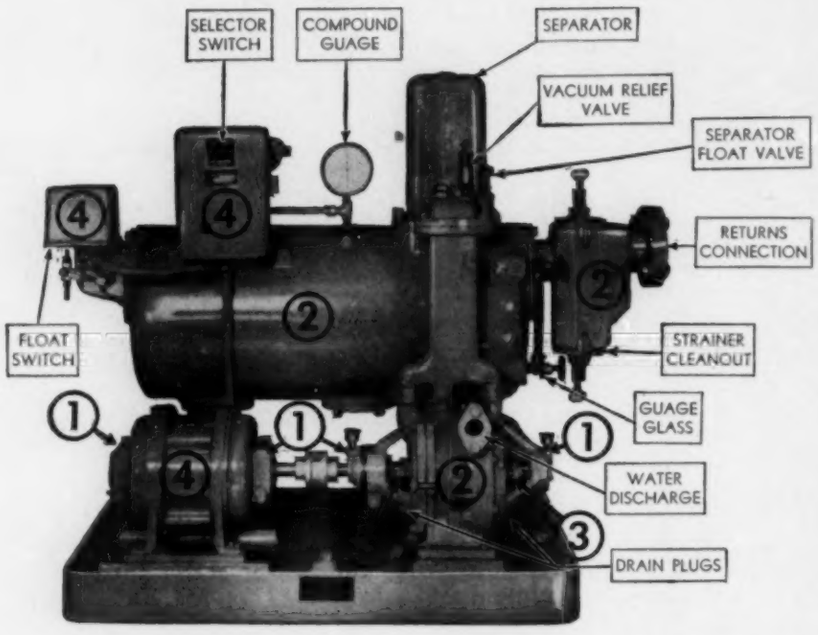
How to keep your Jennings Heating Pumps in First Class operating condition for Duration...



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Ventilation for Control of Fumes in Wartime Welding Work

W. C. L. Hemeon Tells How to Control Electric Welding Fumes to Keep Production Men on the Job in Shipyards and Industrial Plants and Shops

SUMMARY — Control of fumes from electric welding is of immense importance in keeping men on the job, by avoiding the otherwise inevitable days off for sickness and for maintenance of worker efficiency and morale. Necessity for such control is especially apparent in the shipbuilding industry, but is just as essential in all plants and shops employing welding for production for victory. . . . . Mr. Hemeon, engineer with the division of occupational hygiene of the Massachusetts department of labor and industries, gives practical, how-to-do-it data here. He covers local exhaust, the size of the safety zone, and practical limitations, presents a specification, discusses hose size, the possibilities in using static pressure regain, importance of ventilation service crews, use of semi-portable fans, general ventilation, shop ventilation, floor welding, bench welding, and describes a method of measuring fume control

THE NECESSITY for control of fumes from electric welding is most apparent in the shipbuilding industry where thousands of welders are employed, some in the confined spaces of ships under construction, others by the scores in auxiliary shops. Many yard operators, though burdened with the task of getting an important job of production done, have realized that a part of that job is in protecting the welders from excessive concentrations of fumes, although their efforts in that direction have not in all cases been successful.

Control of these fumes is of immense importance for keeping men on the job, by avoiding the otherwise inevitable days off for sickness and for maintenance of worker efficiency and morale. It is unfortunate for the war production program that these factors have not been appreciated in all quarters.

Experience indicates that there is some misapprehension as to the principles of ventilation applicable to this problem, and it is the purpose of this article to deal with these principles and their practical application.

Local Exhaust

For routine control of welding fumes in confined spaces, the practical method is local exhaust, wherein the welder is provided with the free end of a flexible metal hose which leads to an exhaust fan discharging to the outer atmosphere. For welding in ships where location of the work is constantly changing these fans must be portable (with certain exceptions), and there are fans on the market especially developed for this purpose that have a high degree of portability.

The mere presence of the exhaust hose in the compartment with the workman is no guarantee that fumes will be effectively removed. There is nothing automatic about it.

The welder must place the end of the hose within a few inches of the welding point; and since the welding location changes continually, the position of the hose must also be frequently changed. This feature, obviously, is a disadvantage but one which is overcome to a large extent by instructions to the worker and by his own experience in developing the habit. This method of fume control is fundamentally similar to the exhaust units in common use for control of dust in granite cutting, which also requires conscious placing of the exhaust hood near the dust source.

Safety Zone

Air flow through the hose creates a "zone of suction" or "safety zone" around the end of the hose; and if the welding takes place within this zone, all fumes will be effectively removed, otherwise not. The "radius" of this zone is determined by the rate of air flow into the end of the hose and may be calculated in accordance with the equation

$$Q = V\,(10x^2 + A)$$

where Q = cubic feet of air per minute; V = 75 fpm, the minimum velocity *in space* required to capture and direct the fumes into the hose; x = distance in feet, from the end of the hose to the point in space where the velocity of 75 fpm is induced; A = area in sq ft of hose end.

Since V, the controlling velocity for welding fumes, is fixed at 75 fpm [Tebbens and Drinker, Journal of Industrial Hygiene and Toxicology, September, 1941] and A is insignificantly small, the equation may be simplified for welding as follows:

$$Q = 750\,x^2$$

Thus if a "reach" of 12 in. (x = 1) is to be attained, then an air flow of 750 cfm would be required. If the boundary is to be 6 in. distant (x = ½) an air flow of a little less than 200 cfm would be required. An air flow of 50 cfm would

Fig. 1—Illustrating significance of varying air flows through end of hose in terms of size of "safety zone" induced. The larger this zone, the better, but practical considerations limit it. A minimum air flow of 200 cfm is recommended

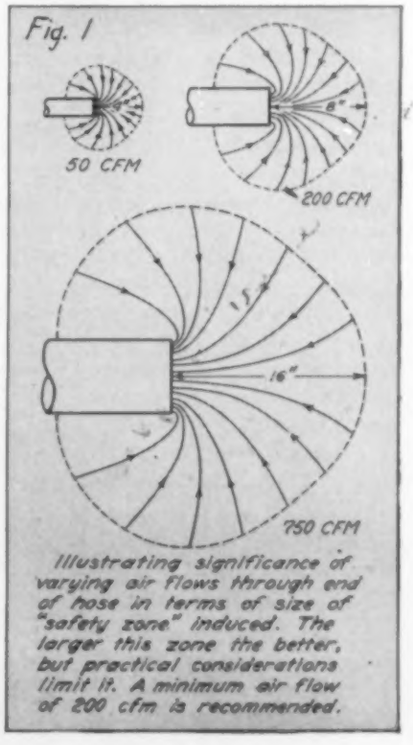


Fig. 1
50 CFM
8"
200 CFM
12"
16"
750 CFM
Illustrating significance of varying air flows through end of hose in terms of size of "safety zone" induced. The larger this zone the better, but practical considerations limit it. A minimum air flow of 200 cfm is recommended.

place the safety zone boundary at a distance of a little over 3 in. ($x = 0.26$ ft.)

The preceding figures apply to the hose and fume source when there is no baffling of air flow in the vicinity. Actually this is almost never the case, because both hose end and source are usually adjacent to a wall which prevents flow of air from that quarter. For this more common situation, the boundary of the safety zone, measured along the axis of the hose end, is about one-third greater than the figures given above. Thus, for an air flow of 50 cfm, the maximum reach is not over 4 in.; for 200 cfm it is about 8 in.; and for 750 cfm, about 16 in. These relations are diagrammatically illustrated in Fig. 1.

Obviously, the larger safety zone created by the larger air flow is best since it requires less attention on the part of the worker to the position of the hose.

Limitations

Air flow and the consequent area of safety zone are, however, limited stringently by practical considerations. Portability requirements fix the maximum size of exhaust fan and weight of motor. The necessity for maneuverability of the flexible hose during use limits its size in practice to 3 or 4 in., with the former limit very much preferred because less cumbersome.

Analysis of the operating characteristics of the portable exhaust fans manufactured primarily for such ventilation work by three manufacturers who specialize in this field, in conjunction with the considerations of safety zone areas and hose maneuverability, justify the following ventilation specification for control of fumes incident to electric welding in confined spaces. (Portable in the sense used here means portable by one man; crane-portable units are discussed later).

Specification

Each welder shall be provided with a flexible metal hose through which is exhausted not less than 200 cfm of air, discharged in such manner that the working atmosphere is not recontaminated.

The above-mentioned commercial portable units will meet this minimum specification when exhausting through approximately 40 ft of flexible hose, a length which corresponds to common practice in shipyards. As shown above, an air flow of 200 cfm provides a safety zone around the end of the hose with an effective radius of about 8 in. (see Fig. 1) which is reasonably adequate, since it avoids the fatal shortcomings of smaller radius zones. That is, too little air flow:

a) requires fussy attention to placement of the end of the hose. Result: Waste motion or incomplete fume control.

b) requires too frequent shifting of the position of the hose. Result: Waste motion or incomplete fume control.

c) renders it impractical in some welding positions to place the hose end close enough to the welding point to remove the fumes. Result: Lack of fume control.

Thus there is a joint responsibility in this method of fume control—the employer's to provide each welder with a safety zone of a minimum size (6 to 8 in. radius as above defined), and the worker's to move the end of the hose as his work progresses so that he is always welding inside the safety zone provided.

Hose Size

In order to demonstrate the capabilities of various portable exhaust units, the pressure-volume characteristic curves for several, together with various system (hose) characteristics, have been plotted, the intersections of which define actual air flow through the hose size or combinations of hoses whose curve is represented. These are shown in Figs. 2 to 6. The pressure loss—volume curves for the hose are shown only for the minimum sizes required to pass 200 cfm.

Friction loss in flexible metal hose was found in field tests to be approximately 25 per cent greater than that in sheet metal ducts, and the system characteristics have been drawn on that basis.

The Roman numerals indicate the number of parallel hose lines represented by a given curve. Curve II, for instance, represents the system resistance of two 40 ft hose lines in parallel and is made up of points representing double the air flow of a single hose line at the same friction loss.

The results of this analysis show that unit "175V" must be used with 4 in. hose to meet the local exhaust specification and that one unit could handle three 4 in. hoses in parallel. This unit is primarily designed for supplying fresh air at low pressure through large diameter canvas duct to spaces requiring it and is therefore not as well adapted to local exhaust.

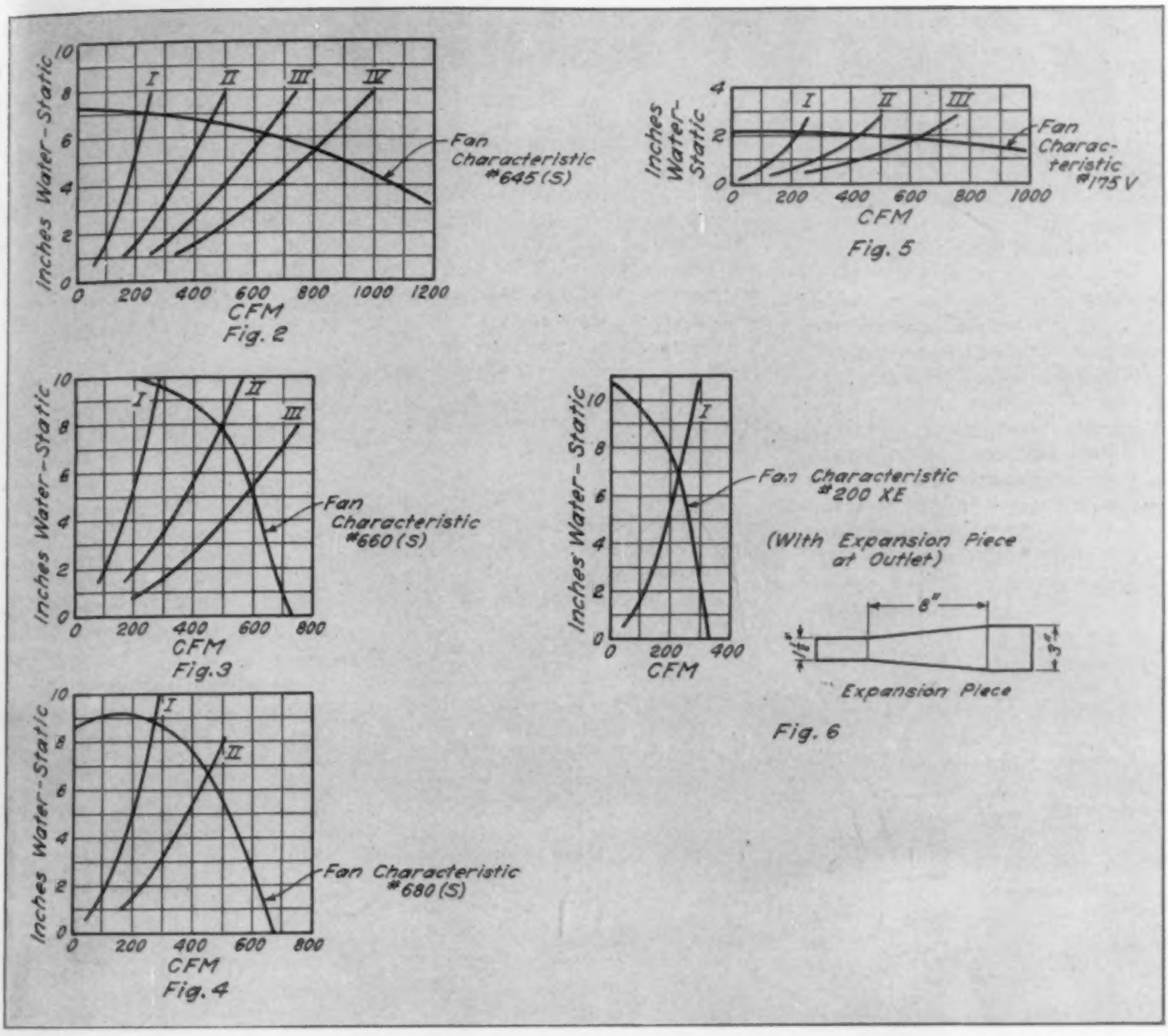
It is seen that units "645 (S)," "660 (S)" and "680 (S)" may be used with 3 in. hose. Unit "645 (S)" can handle four such hoses in parallel, "660 (S)" only three hoses in parallel and "680 (S)," two in parallel.

A comparison among these "(S)" units further indicates that while "645 (S)" is superior to the others for run of the mill work where 40 to 50 ft hose lines are adequate, "660 (S)" and "680 (S)" have greater capacity where necessary to exhaust through greater lengths of hose—up to 75 ft (single line only) for "660 (S)." The capacity of "645" at free delivery is about the same as "175V."

Unit "200 XE" has some interesting features. The analysis of the curves discloses that it will operate in conformity with the 200 cfm specification when used with one 3 in. hose. The inlet and outlet pieces to this fan are 1½ in. diameter, which has led users in shipyards to the erroneous idea that 1½ in. hose should be attached to it. A curve (not shown) for hose of this size discloses a maximum air flow through it of not over 50 cfm, which, as has been shown, is grossly inadequate. Hence, properly used, a 3 in. to 1½ in. tapering connection should be provided at the inlet to accommodate 3 in. hose.

Static Regain

It may be noted, also, that the velocity at the fan outlet of this unit is more than 16,000 fpm (200 cfm, 1½ in. pipe) with a velocity pressure of almost 17 in. water. Here is an opportunity for a really spectacular use of a diverging nozzle, which in ordinary industrial exhaust systems often seems to be worthwhile if it can regain ½ in. static pressure. In the present case a regain of more than 12 in. is easily realized.



Such a fitting can easily be made and attached by the user. The tapering piece 1½ in. by 3 in. diameter should be 8 in. or more long, as shown on Fig. 6. The characteristic curve of Fig. 6 is based on the inclusion of an abrupt expansion, the only data available. It nevertheless serves the purpose of demonstrating that this unit will exhaust well beyond 200 cfm with a pressure regain fitting—even an abrupt expansion. [For a discussion of static pressure regain in air ducts, see article by J. R. Fellows in the April, 1939, HPAC].

Service Crews

The conditions peculiar to shipbuilding that require portable ventilating units are the same ones dictating the necessity for a ventilation crew serving the welders—constantly changing locations, work in spaces not easily accessible, tangle of power lines, hose, etc., and the importance of keeping welders at the job of welding, not pulling hose and fans around from one location to the other. Some large shipyards have such crews, comprising in some cases scores of men whose job is to service the ventilating equipment under the direction of a ventilating engineer. Anyone who has seen the tangle of hose and power lines that results in the absence of such servicing will readily appreciate the truth of the statement that such an organization is of importance equal to proper exhaust equipment.

Semi-Portable Fans

In some circumstances a semi-portable type steel plate fan may be set up with a sheet metal exhaust

Fig. 2—Capacity of unit "645 (S)" is 200 cfm through 40 ft of 3 in. metal hose, maintaining this capacity in each of four hose lines arranged in parallel. Horizontal curve is fan characteristic, while numerals designate hose resistance curves for one, two, three or four lines in parallel. Free delivery capacity is about 1450 cfm

Fig. 3—Fan "660 (S)" has capacity sufficient to serve three 40 ft hose lines in parallel (3 in. diameter), maintaining 200 cfm minimum air flow in each. Curves also indicate it capable of maintaining minimum air flow through 75 ft of a *single hose line*. No. "645 (S)," of Fig. 2 has almost no reserve capacity *as to hose length*

Fig. 4—Fan "680 (S)" has capacity for only two 3 in. hose lines in parallel, 450 cfm for two lines, i.e., 225 cfm through each

Fig. 5—Fan unit "175V" has capacity for three parallel hose lines when hose is 4 in. diameter. Free delivery capacity is about 1500 cfm

Fig. 6—Fan unit "200 XE" with diverging expansion nozzle at outlet (dimensions shown) has capacity of more than 200 cfm when used with a single 40 or 50 ft length of 3 in. hose. For very great lengths of hose, two units may be connected in tandem

pipe extending through hatchways partially into the hull, with 3 in. nipples at convenient intervals to which flexible metal hose may be attached and lead to more distant locations where welding is to be done. As with the portable units, each hose should exhaust at least 200 cfm; hence the fan specification should be based on this quantity times the maximum expected number of hoses to be served. Total pressure will be that through the hose plus additional losses through pipe and will be independent of the number of hoses in use.

Another arrangement sometimes used employs either a steel plate fan or vacuum exhauster sitting on the deck which may be moved by crane as required and which has capacity to serve six or eight hose lines arranged in parallel.

Some yards provide a permanent central exhaust fan with main pipe running the length of the shipway at the side of the hull, with connections for 3 in. hose lines at desired intervals leading across and down into working spaces below. The size of the main pipe should be large enough to result in friction losses that are low compared with those in the branch hose lines. In this way adequate air flows are more readily obtained in branch lines at the tail end of the system. An exhaust fan having a relatively flat (i. e., nearly horizontal) characteristic curve in the operating range has distinct advantages over one with a steep characteristic curve.

General Ventilation

Control of fumes by dilution in confined spaces should not be attempted where production welding of galvanized steel is done on a large scale. General ventilation, therefore, as by means of so-called shiphold ventilators which attempt to discharge large volumes of outside air into the working space should not be considered as other than a temporary measure; for not only is it seldom practical to deliver large enough volumes of air, but it is virtually impossible in practice to avoid kinks in the canvas supply hose which reduce the flow from such portable units markedly. Under favorable circumstances, general ventilation may be satisfactory where welding is on black iron but is not as satisfactory as local exhaust.

Shop Ventilation

The very nature of shop welding, on the other hand, permits greater flexibility, and dilution may in some circumstances be practical in part. (For confined spaces in shops, such as the interior of tanks and pressure vessels, the same standards as described for shipways apply.)

To maintain concentrations of fume from welding of black iron within reasonable limits (15 mgm per cubic meter of air), an air flow of 500 cfm per welder is required where welding rods do not in general exceed 5/32 in. This rod size is the rule in ship work. Higher rates of ventilation are required where larger rods are used or where work is on galvanized metal (see Table 1).

Table 1—General Ventilation Required in Shops for Control of Fumes from Welding*

Rod Size, In.	Ventilation Rate per Welder, cfm
	Black Iron
1/16	500
5/64	800
1/8	1400
5/32	2400
3/16	3000
	Galvanized or stainless
1/8	1500

*Tebbens, B. D., and Drinker, P.: Ventilation in arc welding with coated electrodes. The Journal of Industrial Hygiene and Toxicology, 23, 7, 1941.

It is apparent that where much heavy welding is done or where a large amount of galvanized steel is welded, general ventilation is not an economical method.

Furthermore, in most shops the floor is likely to be very drafty, so that while there is a natural tendency for welding fumes to rise well above the welder's head before dispersing, this is not always the case where there are drafts to blow the fumes toward the welder's head.

Floor Welding

Local exhaust or combinations of this method with general ventilation remain the best methods for most shop welding. Straight local exhaust systems for shops do not require any of the equipment to be portable except the hoses. Therefore, an ordinary centrifugal fan, centrally located, can be used in conjunction with a sheet metal exhaust system having nipple connections at convenient intervals to which flexible metal hose may be attached as desired and run along the floor to welding points.

Where structural columns break the shop into bays, fixed branch lines may be run from an overhead main down such columns to a point near the floor with nipple connections for hoses thus provided at each column.

As in the case of portable units on shipways, local exhaust should provide for an air flow through the hose of not less than 200 cfm. Hence, the fan capacity should provide this air flow through the maximum expected number of hoses to be used, regardless of how many more branch connections with closure caps may be provided for convenience.

When for reason of shop size or special operating difficulty it is not practical to run hose lines from welding points to a system discharging directly outdoors, recourse may be had to portable blowers of the types discussed for shipways with short lengths of hose which serve to remove fume from the vicinity of the welder but which then discharge into the shop atmosphere a short distance away. Supplementing this arrangement, general ventilation is necessary at the minimum rate indicated in Table 1. As has been pointed out, this may mean very wasteful heat loss in cold weather.

Combinations of the above may sometimes be indicated. Thus, for very wide shops, a central station, local exhaust system may serve welding areas adjacent to walls, while those areas in the center which are not convenient to reach from the central station system may be served by portable exhaust units discharging into the shop atmosphere with appropriate amount of general ventilation.

Bench Welding

Where small-sized work is always brought to the same bench location for welding, ideal ventilation may be provided in the form of a local exhaust system. It is common practice to use hood supporting mechanisms of a type originally developed for granite cutting. One of these employs 4 in. rubber hose;

[*Concluded on page 346*]

How a Hotel Chain Maintains Its Air Conditioning Systems

Dinkler System Employs Specialist Who Visits Each Hotel During Winter Months..... Methyl Chloride Substituted for "Freon" in Some Cases Because of War Program ..... By H. W. Genone, Vice-President in Charge of Engineering, Dinkler Hotels

SUMMARY — An organization owning and operating air conditioning installations in several cities must establish certain standards of design, installation, operation and maintenance to assure satisfaction and economy. Planned operating and maintenance procedures are particularly important in wartime because of the greater demands on equipment and material restrictions. . . . . Mr. Genone briefly outlines here some of the practices of the Dinkler chain, which serves seven southern cities. During the winter months, a maintenance specialist from the home office travels from hotel to hotel, inspecting the equipment and making necessary repairs and changes. He is on call 24 hours a day for assistance when anything the local people can't handle develops. . . . . Due to a difficulty in obtaining "Freon" refrigerant because of the war program, two of the installations were changed over to methyl chloride, with satisfactory results.

THE SEVEN Dinkler hotels have approximately 1500 tons of public space air conditioning. Plenty of outside air is used—approximately 15 cfm per person. For example, one of the rooms seats approximately 700 people. It has 100 tons of air conditioning; there are four 25 ton compressors and two 50 ton evaporative condensers. The system is so designed, with pneumatic controls, that every compressor is controlled to take care of the load as the people enter the room. This gives needed flexibility, as sometimes the parties range from 100 to 200 people.

The ventilating system exhausts smoke and air back through the evaporative condensers, which improves performance over using outside air for this purpose.

Bedrooms

Some of the bedrooms of our hotels are served by duct systems, but more recently a method which has proved quite satisfactory during the last two seasons has been adopted. In the Jefferson Davis, at Montgomery, Ala., 28 individual air conditioning units are installed directly under the windows. There is an opening cut through the wall under the window, with a copper screened duct set in flush with the brick wall. In the corner of the room are two circulating lines for chilled water and drain, which run from the third floor to the roof. This piping is boxed in, plastered and covered like a column. The lines are insulated with cork and connected to the shell and tube cooler and circulating pump. The chilled water circulates through the pipe lines, down through the air conditioning unit, and back to the cooler. Operation is economical, for each room is individually air conditioned, which is also of advantage from the standpoint of inter-room noise transmission. As the guest comes into the room, if he desires air conditioning, he turns on his own unit. During the night, if he decides he does not want the air conditioning, he runs just the fan for ventilation. In winter, hot water

Purpose of national hotel week, June 1 to 7, sponsored by the American Hotel Association, is to feature hotels as an asset to the community and a service to the Nation

circulates through the same pipe lines.

Maintenance

For the maintenance of our air conditioning installations, one man travels in winter from hotel to hotel, inspects the machinery, air conditioning units and condensers, and makes necessary repairs and changes. He stays at each hotel and gets each system adjusted before he leaves to go to another. This is a very satisfactory arrangement as the chief engineers and engineers that are stationed at the hotels for other purposes do not understand the fine adjustments of expansion valves and electrical and air controls as well as does an expert. When this man leaves a hotel, he teaches the one who is best fitted for the job to look after the system until he returns. Then, if there is some minor trouble, the local man is able to make repairs. Should serious trouble develop, the maintenance man is on call 24 hours a day for the Dinkler chain.

The standard requirements of the Dinkler chain are based on an inside temperature of approximately 78 F dry bulb and 50 per cent relative humidity and an average of 15 to 20 cfm of outside air per person, depending upon the type of room or space; the amount of smoking governs this to a great extent.

Refrigerant Shortage

The war production program has made it difficult for us, at times, to obtain "Freon" gas, on which all of the air conditioning equipment in the Dinkler hotels has been operated in the past. However, this inconvenience has been overcome when necessary by substituting methyl chloride. Such a change over was made in two particular instances. At one of the "drum" rooms—a drinking place at the Savannah hotel, Savannah, Ga.—a 15 ton unit was changed from "Freon" to methyl chloride and is giving very satisfactory results. At the St. Charles, in New Orleans, methyl chloride was substituted for "Freon" in the 10 ton unit serving the barber shop.

The only trouble caused by changing a system from "Freon" to methyl chloride is in the expansion valves. The present expansion valves have been closed down as much as possible. If these had been permanent change overs, more satisfactory results would be obtained by using expansion valves designed especially for methyl chloride due to the orifice in a "Freon" expansion valve being much larger than for methyl chloride. A factor also to be considered is the effect of methyl chloride on aluminum parts.

Last year it was impossible for us to buy any "Freon" at all; therefore, when the "Freon" on hand was exhausted, the methyl chloride had to be substituted. At the present time, "Freon" can usually be obtained; however, there is a shortage of the cylinders, and cylinders should be returned to the refrigerant manufacturers promptly.

[*Note:* R. J. Thompson, Kinetic Chemicals, Inc., said last month there is enough "Freon" available for servicing all refrigerating machinery in the U. S. and friendly foreign countries, and for the initial charge in such refrigerating machinery permitted to be built under WPB limitation orders.]

Roof Garden

One of our most recent air conditioning installations is for the Ansley roof garden, or Rainbow room, in Atlanta. This room has a very artistic mirror and lighting arrangement carrying out the rainbow effect, with a seating capacity of 700 to 800 people and a dance floor that will accommodate approximately 250. The equipment comprises two 12,000 cfm conditioning units fitted with two circuit cooling coils and heating coils. The refrigeration plant comprises four 25 ton compressors and four evaporative condensers. Zone control, using pneumatic and electric apparatus, is employed.

All four of the 25 ton units are installed for separate operation, this being very desirable for rooms of this type where the load may vary from 25 to 100 per cent of maximum. As a means of adequate ventilation, outside air ducts are installed for maximum fan intake. There is an exhaust system having a maximum capacity of 18,000 cfm, which can be controlled so as to take out air in proportion to load requirements when refrigeration compressors are required, or take out up to 80 per cent of supply fan capacity during mild weather when refrigeration equipment is not needed.

One of the most important air conditioning factors, especially with a room of this type, is the selection of the proper supply outlets and their proper location, so as to have a balanced induction of conditioned air throughout the room.

Ventilation for Control of Fumes—

[*Concluded from page 344*]

others are all metal with 5 in. duct. Air flows of 400 to 500 cfm are commonly provided because these large duct sizes permit them with only moderate friction losses. It may readily be seen that with such air flows the welding safety zone is very large compared with those attainable with portable blowers, and control of fumes is a simple matter.

Measuring Fume Control

A simple suction measurement with a U tube at the free end of the hose provides a convenient check method for plant supervisors, inspectors or others interested in ascertaining to what degree the 200 cfm specification is met. For 3 in. hose the measurement should be made at a small hole approximately 4 to 6 in. from the hose end. A suction of 2 in. of water is equivalent to 200 cfm in hose of this size.

The plant's principal responsibility can be said to have been met if such measurements equal or exceed the above value. Thenceforth, the efficiency of fume control is dependent upon faithful use of these devices by the welders. It is still the plant responsibility, however, to provide proper servicing of the equipment, which includes crews to lay hose lines to welding points from exhausters. In large shipyards such crews ought to be under a man experienced in exhaust systems, for many situations in ship welding call for special combinations of ventilation equipment to effect adequate fume control.

How to Protect Equipment Against Failure of Flow

New Data Allow Easy Selection of Orifices for Use With Differential Pressure Switches to Guard Against Damage from Flow Failure in Chilled Water and Other Piping Systems

By William Goodman

SUMMARY—In war production plants where water is being chilled by refrigeration compressors for process work or air conditioning service, protection against the failure of flow of water is vitally necessary. Failure of flow may cause a disastrous freeze-up. . . . In this article (the first part of which appears this month) the use of orifices and differential pressure electric switches is shown to be a superior method for stopping the compressor in case the flow of water ceases. . . . Although the use of orifices and differential pressure switches for this purpose is not new, nevertheless this type of protection has not been used widely. The principal reason has been due to lack of information in convenient form for quickly and correctly selecting orifices. Therefore, complete data are presented in a readily understandable form for quickly selecting orifices for various conditions of flow and different pressure drops. In addition, the proper method of installing orifices and of connecting them to the pressure switches, together with precautions, are given in a clear and understandable manner. Such data should also be useful in connection with various industrial process piping services other than refrigeration and air conditioning. . . . The author is consulting engineer with the Trane Co., and a member of HPAC's board of consulting and contributing editors

MANY WAR production plants are using chilled water either for air conditioning or for process work. When water is chilled by means of refrigeration, there is always the danger of freezing the water in the direct expansion chiller. Such an occurrence in a plant where continuity of operation is vital is usually little short of catastrophic. The fact that water tubes burst, refrigerant is lost, and that considerable time may elapse before the chiller can be retubed and normal operation resumed may be only a small part of the damage.

Usually a thermostat is installed either in the cold water pipe line on the leaving side of the water cooler or directly through the head of the cooler. In either case, if the temperature falls too low, the thermostat is supposed to stop the compressor. A thermostat in the pipe line is of value only if the water is kept constantly circulating. If the flow of water should cease, freezing could take place in the chiller before the thermostat in the pipe line could stop the compressor. Rapid circulation of the water over the bulb of the thermostat inserted through the head of the cooler is also desirable. In addition to a thermostat, some form of secondary protection is always desirable in case of improper setting of the thermostat or of a mechanical failure. Because damage can occur so quickly if the flow of water ceases, some means of stopping the compressor if this should occur is vitally necessary.

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Various Methods

Two methods of stopping the compressor in case the flow ceases have commonly been used, neither of which offers altogether satisfactory protection. The most widely used is the one in which the refrigeration compressor is interlocked electrically with the chilled water circulating pump. If the pump is not running, the compressor cannot start, or if the circulating pump should stop the compressor will also stop. In the second method a pressure actuated electric switch is installed in the discharge pipe of the water circulating pump. The increase in pressure when the pump starts actuates the pressure switch which then starts the refrigeration compressor. Conversely, the pressure decreases when the pump is stopped, thus stopping the compressor. As is shown in the following two paragraphs, neither electrical interlocking nor a pressure actuated electric switch offers complete protection against a freeze-up due to an interruption of water flow.

Electrical interlocking does not provide satisfactory protection because it is quite possible for a motor not to run even though the current is on. Either low voltage or single phasing may keep current on the motor even though the motor itself is not turning over. In this case, even though the compressor were electrically interlocked with the pump motor, a freeze-up of the water cooler could easily occur.

Although the use of a pressure switch is superior to electrical interlocking, it still has undesirable features. For example, if the water pipe line or the tubes of the water chiller should become obstructed, the pressure would *rise* in spite of the fact that in this case the flow of water would be markedly reduced or even stopped altogether. Such an obstruction could be due to dirt and scale accumulating, or worse still, to freezing in the tubes of the chiller. As freezing started, the water pressure ahead of the chiller would rise as the flow decreased. In fact, if the flow should cease completely, the water pressure at the discharge of the pump would rise to its maximum—that is to the "no delivery" pressure of the pump. In this case, it is obvious that the compressor would continue to run even though the flow had been markedly reduced or even ceased altogether.

Usually the air cooling coil is installed after the water chiller. If a pressure switch is installed between the water cooler and the air cooling coil, stoppage in the piping or the water cooler *ahead* of the pressure switch would cause a drop in pressure. In this case, the pressure switch would stop the compressor. However, stoppage anywhere beyond the pressure switch would cause an increase in pressure even though the flow had ceased completely.

Orifice and Switch

A most satisfactory method of protecting a chilled water circulating

system against danger of a freeze-up is by means of a unit consisting of an orifice and a differential pressure switch. With such a unit, protection can be obtained not only against complete failure of water flow, but also against the possibility of the water flow dropping to any predetermined dangerously low value. (A freeze-up can also occur if the quantity of water circulated decreases to such a low point that the water is chilled excessively.)

If the flow through an orifice decreases, the pressure drop across the orifice also decreases. Thus if the flow through an orifice were decreased from full flow to, say, 50 per cent, the pressure drop across the orifice would be decreased to 25 per cent of the original pressure drop with full flow. This change in pressure drop can be used to actuate a differential pressure switch which in turn can be used to stop the compressor, close a solenoid valve in the refrigerant liquid line or operate any other device. Of all the safety controls available, the orifice and differential pressure switch unit alone has the unique advantage of offering protection not only against complete failure of flow, but also against a reduction in flow down to any desired predetermined value.

The differential pressure switch should not be installed without an orifice. It is tempting to utilize the drop in pressure across the equipment rather than the drop in pressure across an orifice. For example, the difference in pressure between the inlet and outlet of the water chiller could also actuate a differential pressure switch because this difference in pressure changes with the quantity of water flowing through the cooler. However, an obstruction in the tubes due to dirt or due to the beginning of freezing would also cause an increase in pressure difference even though the water flow might be reduced considerably. In this case it is obvious that the differential pressure switch could not stop the compressor.

Maximum Drop

Differential pressure switches are actuated only by the difference in pressure in two bellows or on two sides of a diaphragm. The absolute value of the pressure in either bellows or on either side of the diaphragm has no effect on the operation of a differential switch. Inasmuch as the difference in pressure across an orifice depends upon the amount of liquid flowing through the orifice, it is apparent that a change in flow will cause a change in the pressure drop across the orifice which in turn can be used to actuate a differential pressure switch.

Orifices and differential pressure switches must be selected and designed for the system to which they are to be applied. Inasmuch as an orifice introduces a permanent pressure loss into the system, the orifice must be selected for as small a pressure drop as possible. The *permanent pressure loss* should not be confused with the *pressure drop* across the orifice. As is shown later, the pressure loss is always less than the pressure drop. However, the size of the pressure loss depends upon the size of the drop. This permanent pressure loss can be kept so low that it will not be important insofar as the operating cost is concerned. How low this permanent pressure loss can be kept depends only upon the minimum differential which will operate the pressure switch.

Suppose, for example, that with a flow of 100 gpm, the pressure drop across an orifice is 8 psi [psi means "pounds per square inch"]. If the flow drops to 50 per cent of this value, or 50 gpm, the pressure across the orifice will fall to 2 psi because the pressure drop varies as the square of the flow. The difference in pressure drops between the maximum and minimum flows is 6 psi. In other words, the differential pressure switch must be capable of cutting in and cutting out on this 6 psi difference.

Differential pressure switches which will operate on a smaller difference in pressure drop than 6 psi can be obtained. The actual difference usually depends on the total absolute pressure to which the system is subjected. Thus, one manufacturer lists pressure switches which can withstand total pressures up to 85 psi. These switches will operate on a minimum difference in pressure drops of 1½ psi. Pressure switches which can withstand a total absolute pressure of only 20 psi can be operated on a difference in pressure drops of only ¾ psi. For the usual system, it is probably best to choose the heavier switch. In selecting an orifice, it is best to choose a difference in pressure drops at least a pound greater than the minimum for which the differential pressure switch is rated. This will provide a slight margin of safety

Symbols

A_o = orifice area, sq ft;
d = difference in pressure drops for maximum and minimum flow, psi;
D_o = orifice diameter, in.;
D_p = inside diameter of pipe, in.;
g = gravity constant, 32.17 ft per sec per sec;
h_w = head, ft of water;
K = flow coefficient of orifice;
n = ratio of minimum to maximum flow;
p = pressure drop across orifice for maximum flow, psi;
p_1 = pressure drop across orifice for exact sizes tabulated in Table 2, or for less than maximum flow, psi;
p_L = permanent pressure loss across orifice, psi;
Q = maximum flow, gpm;
Q_1 = flow, less than maximum, gpm;
Q_o = flow, cu ft per sec;
s = specific gravity of fluid (for water = 1);
Z = orifice constant;
Z_1 = orifice constant for pressure drop p_1.

TABLE 1

MAXIMUM PRESSURE DROP ACROSS ORIFICE FOR MAXIMUM FLOW

Pounds per Square Inch (Psi)

Difference in Pressure Drops, Psi	Minimum Flow in Percent of Maximum Flow						
	30%	40%	50%	60%	70%	80%	90%
0.25	0.3	0.3	0.3	0.4	0.5	0.7	1.3
0.50	0.5	0.6	0.7	0.8	1.0	1.4	2.6
0.75	0.8	0.9	1.0	1.2	1.5	2.1	3.9
1.0	1.1	1.2	1.3	1.6	2.0	2.8	5.3
1.5	1.6	1.8	2.0	2.3	2.9	4.2	7.9
2.0	2.2	2.4	2.7	3.1	3.9	5.6	10.5
2.5	2.7	3.0	3.3	3.9	4.9	6.9	13.2
3.0	3.3	3.6	4.0	4.7	5.9	8.3	15.8
4	4.4	4.8	5.3	6.2	7.8	11.1	21.1
5	5.5	6.0	6.7	7.8	9.8	13.9	26.3
6	6.6	7.1	8.0	9.4	11.8	16.7	31.6
8	8.8	9.5	10.7	12.5	15.7	22.2	42.1
10	11.0	11.9	13.3	15.6	19.6	27.8	52.6
12	13.2	14.3	16.0	18.8	23.5	33.3	63.2
14	15.4	16.7	18.7	21.9	27.5	38.9	73.7
16	17.6	19.0	21.3	25.0	31.4	44.4	84.2

in case the maximum flow is smaller than anticipated. For the usual case, a design difference in pressure drops of 2½ or 3 psi will be satisfactory although other drops can of course be chosen.

The minimum difference in pressure drops available to actuate the pressure switch together with the allowable reduction in flow determine the required pressure drop for the *maximum* flow through the orifice. The pressure drop corresponding to the *maximum flow* will be called for convenience the *maximum pressure drop*.

The maximum value of the pressure drop is needed properly to size the orifice. The maximum pressure drop can be determined conveniently from Table 1. Table 1 was calculated by means of the following equation derived in the footnote.[1]

$$p = \frac{d}{1 - n^2} \quad \text{.....[1]}$$

The use of Table 1 is illustrated by example 1:

Example 1—Determine the maximum pressure drop that will be required if the minimum difference in pressure drops required to operate the differential pressure switch is to be 3 psi. The pressure switch is to stop the compressor if the flow falls to 50 per cent of its maximum value.

Solution—Run down the first column of Table 1 to a pressure drop of 3 psi. Opposite this value in the column for a 50 per cent reduction in flow read 4.0 psi as the maximum pressure drop required for the maximum flow.

From example 1, it is apparent that when the flow is at a maximum, the pressure drop across the orifice will be 4.0 psi. When the flow falls to 50 per cent of its maximum value, the pressure drop will be 1.0 psi. The difference in pressure drops corresponding to the maximum and minimum flows is 3 psi.

TABLE 2

VALUES OF ORIFICE CONSTANT "Z"

Orifice Diameter Inches D_o		Nominal Pipe Size — Inches								
		2	2-1/2	3	4	5	6	8	10	12
		Inside Diameter — Inches								
		2.067	2.469	3.068	4.026	5.047	6.065	8.071	10.192	12.090
5/8	0.625	7.0								
3/4	0.750	10.2	10.1							
7/8	0.875	14.0	13.8							
1	1.000	18.6	18.2	18.0						
1-1/8	1.125	24.0	23.3	22.9						
1-1/4	1.250	30.6	29.2	28.5	28.0					
1-3/8	1.375	38.4	36.1	34.8	34.1					
1-1/2	1.500		44.2	41.8	40.8	40.2				
1-5/8	1.625		53.4	49.8	48.1	47.4				
1-3/4	1.750		64.3	58.9	56.2	55.2				
1-7/8	1.875			69.1	65.0	63.6				
2	2.000			80.6	74.5	72.7	71.9			
2-1/8	2.125			93.8	85.1	82.5	81.4			
2-1/4	2.250				96.8	93.1	91.6			
2-3/8	2.375				109.6	104.3	102.5			
2-1/2	2.500				123.5	116.4	114.0	112.1		
2-5/8	2.625				138.9	129.4	126.3	123.8		
2-3/4	2.750				155.8	143.6	139.3	136.2		
2-7/8	2.875					158.9	153.0	149.2		
3	3.000					175.2	167.6	162.9	161.0	
3-1/4	3.250					211.7	199.8	192.3	189.6	
3-1/2	3.500					254.4	236.3	224.6	220.8	
3-3/4	3.750						277.5	259.7	254.4	252.1
4	4.000						324.1	298.1	290.8	287.8
4-1/4	4.250						377.3	340.4	329.8	325.9
4-1/2	4.500							387.0	371.9	366.5
4-3/4	4.750							438.0	416.7	410.0
5	5.000							493.9	465.1	456.2
5-1/2	5.500							622.1	573.2	557.3
6	6.000								698.9	670.6
6-1/2	6.500								843.9	800.0
7	7.000								1012.4	946.3
7-1/2	7.500									1111.3
8	8.000									1298.4
8-1/2	8.500									1512.2

Selecting Orifice

To select an orifice, the following formula is used:

$$Z = Q\sqrt{\frac{s}{p}} \quad \text{.....[2]}$$

After computing the value of the orifice constant Z from equation 2, the proper orifice size for any given pipe size can be selected from Table 2. The derivation of equation 2, together with the method of calculating the values of Z, is given in the footnote.[2] Example 2 illustrates the method of selecting an orifice.

Example 2—An orifice is to be installed in a 3 in. standard weight steel pipe through which chilled water is flowing. The maximum flow is 112 gpm. The difference in pressure drops for actuating the differential pressure switch is not to exceed 2 psi. The differential pressure switch is to stop the compressor when the flow drops to 60 per cent of the maximum value. Find (a) the maximum pressure drop; and (b) the size of orifice required.

Solution—(a) From Table 1, for a difference of 2 psi and a flow reduction to 60 per cent, $p = 3.1$ psi. (b) Using equation 2 and substituting the values of $Q = 112$, $p = 3.1$ psi, and $s = 1$,

$$Z = 112\sqrt{\frac{1}{3.1}} = 63.6$$

Referring to Table 2 in the column for a 3 in. pipe, the orifice size correspond-

[1]Derivation of equation 1: The pressure drop through an orifice varies as the square of the flow. Hence,

$$\frac{p_1}{p} = \left(\frac{Q_1}{Q}\right)^2$$

or

$$\frac{p_1}{p} = n^2$$

Also,

$$p - p_1 = d$$

Eliminating p_1,

$$p = \frac{d}{1 - n^2}$$

[2]Derivation of equation 2:—The basic equation for flow through an orifice is

$$Q_o = KA_o\sqrt{2gh_w}$$

Changing units, this equation reduces to

$$Q = 29.84\,KD_o^2\sqrt{\frac{p}{s}}$$

Let $Z = 29.84\,KD_o^2$

Hence,

$$Z = Q\sqrt{\frac{s}{p}}$$

The values of Z in Table 2 were computed from the third equation above. The values of K for radius taps were read from Figs. 36a to 36d inclusive of the ASME publication Flow Measurement—1940. The values of K used in preparing Table 2 were all read for a constant Reynolds number of 200,000. Although the value of K does decrease with increasing values of the Reynolds number, the decrease is small and is not of much importance insofar as the selection of orifices for protection against flow failure is concerned.

Regardless of the pipe size, the values of K are substantially constant for given values of the ratio D_o/D_p. Accordingly a curve was prepared for the averaged values of K versus this ratio. The values of K required in the third equation above were read from this curve.

ing to a value of $Z = 63.6$ is found to lie between 1¾ and 1⅞ in. By interpolation, the correct orifice size is 1.81 in.

As a matter of fact, there is no need to resort to interpolation when using Table 2. As far as practical considerations go, it would be quite satisfactory to select the orifice size corresponding to the next *smaller* value of Z. Thus, for $Z = 63.6$, the next smaller value of Z listed in Table 2 for 3 in. pipe is 58.9. The orifice size of 1¾ in. corresponding to this latter value of Z would be quite satisfactory for this purpose. With the orifice size slightly smaller, the difference in pressure drops between the maximum and minimum flows will be slightly greater than 2 psi. By selecting a slightly smaller orifice, a slight margin of safety is obtained.

If desired, the actual pressure drop that will take place with a slightly smaller orifice can be computed by means of the following equation.

$$p_1 = p \left(\frac{Z}{Z_1} \right)^2 \quad \text{.....[3]}$$

Example 3—Find the maximum pressure drop that will take place for the conditions of example 2 if an orifice of 1¾ in. is selected instead of the more exact size of 1.81 in.

Solution—From example 2, $p = 3.1$ psi and $Z = 63.6$. From Table 2, for a 1¾ in. diameter orifice and a 3 in. pipe $Z_1 = 58.9$. Substituting these values in equation 3,

$$p_1 = 3.1 \left(\frac{63.6}{58.9} \right)^2$$

$= 3.6$ psi, actual pressure drop through the 1¾ in. orifice at the maximum flow of 112 gpm.

In selecting the orifice size from the table, the orifice corresponding to the next smaller value of Z should be selected as was done in the preceding example. If an orifice corresponding to the next larger value of Z were selected, the difference between the pressure drops at the maximum and minimum flows might be too small to insure proper operation of the differential pressure switch.

Where the value of Z computed by means of equation 2 is larger than the largest value shown in Table 2 for the given pipe size, either a larger difference in pressure drops must be used or the orifice must be installed in a larger size pipe. In the latter case, a larger pipe need be used for only a short length before and after the orifice. The minimum length of larger pipe required can be found by means of tables to be published with the next installment of this article.

TABLE 3

PRESSURE LOSS ACROSS ORIFICE IN PERCENT OF PRESSURE DROP

Orifice Diameter Inches D_o	Nominal Pipe Size — Inches								
	2	2-1/2	3	4	5	6	8	10	12
5/8	89.6%								
3/4	85.2	89.5%							
7/8	80.2	85.8							
1	74.5%	81.8%	88.0%						
1-1/8	68.3	77.2	84.9						
1-1/4	61.2	72.4	81.6	89.1%					
1-3/8	53.6	66.9	77.9	86.8					
1-1/2		61.8%	74.0%	84.3%	90.0%				
1-5/8		54.5	69.8	81.8	88.2				
1-3/4		47.8	65.4	79.0	86.4				
1-7/8			60.5	76.2	84.5				
2			55.2%	73.4%	82.5%	87.7%			
2-1/8			50.0	70.1	80.4	86.2			
2-1/4				66.6	78.1	84.4			
2-3/8				63.0	75.8	82.8			
2-1/2				59.2%	73.4%	81.2%	89.1%		
2-5/8				55.2	71.0	79.2	88.1		
2-3/4				51.2	68.2	77.5	86.8		
2-7/8					65.4	75.5	85.7		
3					62.5%	73.4%	84.3%	90.1%	
3-1/4					56.2	69.2	81.8	88.5	
3-1/2					50.0	64.5	79.2	86.7	
3-3/4						59.5	76.4	84.8	89.1%
4						54.2%	73.3%	82.8%	87.6%
4-1/4						48.8	70.2	80.7	86.0
4-1/2							66.8	78.5	84.4
4-3/4							63.2	76.2	82.8
5							59.3%	73.8%	80.9%
5-1/2							51.5	68.2	77.2
6								63.2	73.3
6-1/2								57.0	69.0
7								51.8%	64.2%
7-1/2									59.3
8									54.0
8-1/2									48.5

Pressure Loss

The entire maximum pressure drop across an orifice need not be added to the total head against which the circulating pump is working. A portion of the velocity head of the liquid flowing through the orifice is recovered and hence the pressure loss is less than the maximum pressure drop through the orifice. It is important to remember that only the *permanent pressure loss* of the orifice is added to the pump head, not the *pressure drop* across the orifice. The *permanent pressure loss* of the orifice can be computed by means of the values in Table 3. For the method of computing Table 3, see the footnote.[3]

The use of Table 3 is illustrated by example 4.

Example 4—Take the conditions of example 3 and compute the pressure loss due to the orifice.

Solution—For a 1¾ in. orifice in a 3 in. pipe, find in Table 3 that the pressure loss will be 65.4 per cent of the maximum pressure drop. From example 3, maximum pressure drop = 3.6 psi. Hence, $p_L = 0.654 \times 3.6 = 2.4$ psi permanent pressure loss due to the orifice. When selecting a pump, the permanent pressure loss through the orifice of 2.4 psi must be added to the total head for which the pump is to be selected.

[Mr. Goodman's next article on this subject will cover construction and installation of orifices, pressure tappings, connections to the pressure switch, manometers, and liquids other than water.—Ed.]

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[3]Computation of the values of permanent pressure loss in Table 3:—According to paragraph 56 on p. 17 of the ASME publication Flow Measurement—1940, the permanent pressure loss may be computed approximately from the formula:

$$\frac{p_L}{p} = \frac{1 - K\left(\frac{D_o}{D_p} \right)^2}{1 + K\left(\frac{D_o}{D_p} \right)^2}$$

As mentioned in footnote 2, average values of K corresponding to different values of the ratio (D_o/D_p) were determined. Substituting these corresponding values of K and (D_o/D_p) in the equation above, it was possible to determine different values of the ratio (p_L/p) corresponding to values of the ratio (D_o/D_p). A curve was then plotted of (p_L/p) versus (D_o/D_p). Values of (p_L/p) read from this curve were used in preparing Table 3.

CODING CHART SIMPLIFIES HANDLING OF PD-1A FORMS

SHOWN BELOW is a coding chart for air conditioning and refrigerating equipment. This form was prepared by the Air Conditioning & Refrigerating Machinery Association with the informal collaboration of officials of the air conditioning and commercial refrigeration branch of the War Production Board, who have expressed the hope that members of the industry will make general use of the coding chart, according to William B. Henderson of the ACRMA.

To illustrate how the chart should be used: A copy should be attached to each PD-1A form going to the air conditioning and commercial refrigeration branch in the War Production Board. If the PD-1A form covers, say, the sale of air conditioning equipment for the gage room of a company making machine tools, the coding chart may be marked by checking:

No. 21 in the column entitled "Ultimate Buyer or User of Facility or End Product"

No. 17 in the column entitled "Manufacturing Industry or Business Classification"

No. 9 in the column entitled "Application or Functional Use of Refrigerating and/or Air Conditioning Equipment"

By referring to the coding chart, properly marked, the branch staff can easily allot the proper rating to the application. In this way, PD-1A forms may be cleared by the branch much more quickly.

This coding chart may be duplicated, but *should be sent to WPB only on a size 8½ in. x 11 in. sheet.*

Coding Chart for Air Conditioning and Refrigeration Applications

Ultimate Buyer or User of Facility or End Product	Manufacturing Industry or Business Classification	Application or Functional Use of Refrigerating and/or Air Conditioning Equipment
1. Army Air Corps.	*Manufacturing Industries*	1. Preserving foodstuffs in storage.
2. Army Engineers.	1. Aircraft, including engines.	2. Preserving foodstuffs in transit.
3. Army Ordnance.	2. Automobiles, trucks, and tanks.	3. Cooling or freezing foodstuffs during processing.
4. Army—Other Departments.	3. Chemicals and explosives.	4. Preservation or conditioning materials or products other than foodstuffs in storage or transit.
5. Navy—Bureau of Ships.	4. Communication equipment.	5. Cooling, freezing, or conditioning materials during process of manufacture, including ice manufacturing.
6. Navy—Bureau of Yards and Docks.	5. Drugs and pharmaceuticals.	6. Conditioning spaces in which manufacturing processes are being carried out and/or industrial products inspected or tested.
7. Navy—Bureau of Ordnance.	6. Food—bakery products.	7. Conditioning spaces other than manufacturing spaces, for health, efficiency, or comfort of occupants.
8. Navy—Other Bureaus.	7. Food—beverages.	8. Cooling drinking water.
9. Maritime Commission.	8. Food—dairy products.	9. Conditioning spaces in which fine instruments, gages, and other sensitive apparatus are stored or used.
10. Other U. S. Government Agencies.	9. Food—eggs and poultry.	10. Cooling or freezing materials or conditioning spaces, simulating unusual conditions in research laboratories.
11. Lend-Lease.	10. Food—fish.	11. Cooling water and liquids where controlled temperature is necessary to the production of the product(s).
12. Export.	11. Food—fruit and vegetables.	12. Repairs, replacements, maintenance, and supplies.
13. State and Municipal.	12. Food—meat packing.	13. Other applications (explain in detail).
14. Other Orders Rated AA to A-1-L.	13. Food—quick freezing.	
*15. Dairy Products—Storage and Transportation.	14. Food—other foods.	
*16. Food—Storage and Transportation.	15. Ice manufacture and storage.	
*17. Food and Dairy Product Processing.	16. Iron and steel production.	
*18. Ice Manufacture and Storage.	17. Machinery and metal working.	
*19. Retail Food and Dairy Product Establishments.	18. Non-ferrous metal production.	
*20. Industrial Processing—Refrigerating Equipment.	19. Paper, paper products, printing and lithographing.	
*21. Industrial Processing—Atmospheric Control.	20. Petroleum and coal products.	
*22. Industrial Liquid Cooling, Including Drinking Water.	21. Precision instruments.	
23. Repair and Maintenance Orders.	22. Rubber and rubber products.	
24. All Others.	23. Textiles.	
*Orders Rated A-2 through A-9.	24. All other manufacturers.	
	Businesses	
	25. Cold storage warehousing.	
	26. Hospitals and institutions.	
	27. Housing—auditoriums and theaters.	
	28. Housing—banks and offices.	
	29. Housing—hotels and apartments.	
	30. Housing—residences and air-raid shelters.	
	31. Laboratories.	
	32. Restaurants.	
	33. Stores—food—retail.	
	34. Stores—others—retail.	
	35. Transportation—marine.	
	36. Transportation—railroad.	
	37. Transportation—truck.	
	38. All others.	

WATER TREATMENT CONSERVES HEATING AND PIPING EQUIPMENT

IT IS always good business to operate and maintain mechanical equipment so that it will perform efficiently for as long a time as possible. In wartime, with restrictions on materials and tremendous demands for mechanical equipment for production for victory, it becomes a patriotic duty as well. Proper water treatment and corrosion control is one method of making your present heating and piping equipment last longer and operate at maximum economy.

Different waters in different equipment present varying problems. In general, however, scale formation in a boiler is caused by precipitation of calcium and magnesium salts upon the metals in the boiler. These scale formations are detrimental because they retard the flow of heat from the metallic surface to the water. This has the ill

effects of increasing the amount of fuel necessary to heat the same volume of water to the same temperature, and of causing the tube to burn by not taking away from it rapidly enough the heat imparted to it by the flame.

In heating boilers where the return is wasted and the feedwater has a high hardness, only very short operation can be experienced before it becomes necessary to shut down and clean the boiler to prevent damage. For example, before treatment, a certain 100 hp HRT could operate only one week to 10 days before it became necessary to shut down and remove almost ⅛ in. of hard scale. After proper treatment, this same boiler operated safely for six week periods and after shutting down, required only a hose to remove a small amount of soft mud remaining in the boiler. The resulting fuel saving and increased efficiency permitted only two boilers to carry the load formerly carried by four. When contemplating expansion of existing plant facilities, do not fail to consider increasing the efficiency of present equipment. Perhaps two boilers are doing the work of one.

The most efficient heating systems are those with high percentages of return. Since such a system uses relatively small quantities of raw water, scale formation is minimized. The average operator experiences no difficulty in firing the boiler and does not have to clean out much scale when and if he washes his boiler. This leads to the false assumption that the equipment is and will remain in good condition. To a greater or lesser degree, corrosion is taking place within the system and the steam lines. This is caused by the acidic nature of the water accelerated by the presence of dissolved CO_2 and oxygen. In steam lines, corrosion is caused by the dissolved gases—CO_2 and O_2.

Illustrative of the damage caused by corrosion is the case of a 100 hp boiler which required at least 15 new tubes each heating season. By maintaining a hardness below 1 grain per gal, a pH value of 11.0, an NaOH and Na_2CO_3 alkalinity of 250 ppm, and zero O_2 content, no tubes have been replaced for over 11 years. The average cost of treatment was $45 yearly over this period. This can be compared with the cost of installation of 15 tubes, together with the inconvenience and time loss involved. Such replacements at present would be prohibitive.

Treatment of a heating system has in the past few years been expanded to include conditioning of steam lines as well. A volatile amine which goes off with the steam to render both the steam and condensate alkaline instead of acidic is one such method. Although this compound has a slight alkalinity by itself, it acts more as a carrier and must be regenerated by an alkaline boiler water after having passed through steam lines and having reacted with the CO_2 therein.

A large central heating plant using four 150 HRT's and a 200 hp water tube boiler replaced an average of 100 ft of return line yearly. Since starting treatment 10 years ago, no lines have been replaced. The volatile compound has been used only for the last few years, but contributed measurably to the longevity of the equipment; previous corrosion inhibition was due mainly to the effective removal of dissolved oxygen. Analyses show that a pH value of 8.5-9.0 in the return condensate can readily be obtained by the use of this compound.

In a 500 hp return tubular boiler in an industrial plant engaged in drying lumber, this volatile corrosion inhibitor has been successfully used for over a year. The system has a 45 per cent return and maintained a condensate pH previous to treating of 6.7. By the addition of 1/10 lb of the inhibitor per day, the pH value was raised above 7.1. Concentrations of 0.05 to 0.1 per cent are sufficient to provide properly conditioned condensates for most heating systems.

Many times by ignorance—often by negligence—boilers become deeply pitted and rusted while they are idle during the summer months. A thoroughly dry atmosphere would make the opening of boilers quite satisfactory, but due to the locations of most boilers, this is not the case. Condensation of moisture on the interior of the boiler due to temperature change deposits a water high in both dissolved CO_2 and O_2, causing accelerated corrosion and pitting. To prevent this the boiler may be filled with clear water to which is added a pickling compound which contains inhibitors as well as CO_2 and O_2 removers.—J. Neil Smith, Jr., chief chemist, Anderson-Stolz Corp.

SALVAGING SCRAP TO WIN THE WAR

"There is a new slant today on the necessity of keeping scrap losses down. Before priorities, salvage was entirely on a cost basis. The amount of money saved decided how much was spent on salvage. Now, it's different. Cost, in many cases, is held secondary. Conservation of materials is a matter of self-preservation. In all our plants we have consistently hammered the point home to our employees that our very jobs depend on efficient usage of time, materials, and tools.

"In many cases, we are being forced to substitute more expensive materials, and this boosts unit cost. Reduction of waste and efficient salvage thus become even more important; they help maintain unit cost by partly balancing the expense of substitution, and the manufacturer is better able to meet competition prices.

"It is said that salvage programs are fine for large plants but aren't practical in small ones. Large or small, there are opportunities for salvage that are overlooked. Last summer, I was in one shop after another where ferrous and non-ferrous materials were badly mixed. Tool steel and nickel bearing steels were mixed with machine shop turnings and borings. The man who buys these discards will pay only for the gross weight at the unit price of the most inferior metal in the lot.

"The organization of the plan to conserve and reclaim materials starts with design and ends with disposition of scrap." — Ray Schmidt, supervisor, reclamation and salvage departments, Westinghouse Electric & Mfg. Co., in a talk before the Cleveland Chamber of Commerce.

PLASTIC TUBING — A "War Baby"

Here's Information on Properties and Handling of a New Material With Many Possibilities in Heating, Piping and Air Conditioning Work . . . . . By C. B. Branch and D. L. Gibb

SUMMARY — "Alternative" or "substitute" materials (as you prefer) comprise one of the most talked of subjects whenever heating, piping and air conditioning men congregate. Wartime scarcities and consequent restrictions have focussed the spotlight of attention on many new products which perhaps may be called "war babies" . . . One of the most interesting of these is plastic tubing, available in a number of types. C. B. Branch and D. L. Gibb, of the plastic sales division of Dow Chemical Co., describe here the properties and applications of one of the plastic tubings. They include a listing of characteristics which will enable you to judge its uses, and provide data on strength, effect of temperature, and how to flare and bend it.

IT HAS been said that the machine age began with the invention of the wheel, but certainly the wheel is little more essential to the whole structure of modern life than the tube. Man depends on some form of hollow conductor for the delivery of food and drink, for heat, light, transportation and sanitation. It is not strange, then, that a reduction in the amount of tubing available should present serious problems to innumerable persons and industries; nor that the new plastic tubings are of general interest. One such tubing is made of a thermoplastic formed by the copolymerization of vinylidene and vinyl chlorides. The basic polymer is tasteless, odorless, nontoxic and nonflammable.

This tubing is being produced commercially by a number of licensed fabricators throughout the country. It is made in seamless, continuous lengths, and at present is available in sizes from ⅛ to ¾ in. diameter, with varying wall thicknesses. Present trends point to the fabrication of even larger tubing in the near future.

This plastic is particularly suited for tubing because of its relatively sharp melting point and resultant retention of physical properties within the service temperature range. Another advantage is its comparatively high bursting pressure. Some individual values on ¼ in. OD by 0.037 in. wall thickness at 77 F have been as great as 2250 psi. When the temperature is decreased from 77 F to 32 F, the working pressure is approximately doubled; at the same time, this change in bursting pressure is accompanied by a corresponding reduction in impact strength.

The accompanying table shows the changes in bursting pressure with changes in cross sectional area. It will be noted that the operating pressures for most tubing installations are frequently well within the listed pressure limits.

The following listing of service characteristics indicates the advantages and limitations of this plastic tubing. The chemical resistance tests were based on three-month tests at room temperature:

Acids	Excellent
Caustic	Good
Bleaching agents	Excellent
Salts	Excellent
Ammonia	Poor
Solvents	Good*
Water	Excellent
Air	Excellent
Oil	O.K.
Low Temperature	Becomes stiff and brittle
70 F—170 F	Flexible
High temperature	Not recommended over 170 F
Flexing	Excellent
Freezing cycles, closed system	Excellent
Vibration fatigue	Excellent
Heat conductivity	Slow
Fluid flow	Equal to copper
Physical state	Flexible
Clarity	Semi-transparent

*Not recommended for certain high oxygen bearing oxides and others and some chlorinated aromatic hydrocarbons.

This tubing may be joined by some of the same types of connectors commonly employed with metal tubing. These include flared, compression, welded and flanged fittings. For the smaller sized tubings now available, flared types are very satisfactory. The flaring can be accomplished at room temperature with compression type flaring tools, or by spinning. If the tubing is cold, it is recommended that the end of the tubing and the flaring tool be heated to room temperature before flaring is attempted. [*Note:* In an experimental setup by an air conditioning contractor, the tubing was heated to 100 F by dipping it into warm water before flaring, and this was found to give good results. —Ed.] Any necessary trimming may be done with a sharp cutting tool or file.

Flare fittings made from injection molded plastic are also available. These fittings are slight modifications of the standard SAE fitting designs, and have the advantage of providing a plastic seal between the tubing and the fitting. This combination eliminates possibility of leakage due to unequal thermal expansion of tubing and fittings and provides for uniform chemical resistance.

In bending the tubing at room temperature, the tendency to kink decreases with increasing wall thickness. At a wall thickness equal to 20 per cent or more of the diameter of the tubing, it is extremely difficult to kink or collapse. Even in those cases where actual kinking or total collapse occurs, the bursting pressure is reduced no more than 20 per cent. Fabrication such as bending or forming to permanent desired

Bursting Pressures of Tubing at 77 F

OD In.	Wall, In.	Bursting Pressure, Psi	Working Pressure, Psi
⅛	0.031	1900	380
3/16	0.031	1250	250
3/16	0.031	900	180
¼	0.045	1350	270
¼	0.062	1900	380
5/16	0.031	600	120
5/16	0.062	1700	340
⅜	0.031	575	115
⅜	0.062	1250	250

How the plastic tubing described in text is flared. From top to bottom—Trimming the tubing for the flaring operation . . . Inserting tubing in flaring tool . . . Applying pressure to flare the tubing . . . Finishing the flare.

shapes is easily accomplished. The tubing is formed at room temperature to the required shape plus allowance for 20 to 30 per cent springback. It is then heated to approximately 212 F for 15 to 30 sec and on cooling will be set to shape. The heating in this procedure may be easily and conveniently obtained with boiling water or atmospheric steam. Without internal support, this tubing can be permanently bent without kinking on a radius as small as three to six times the diameter of the tubing. Bends can be made to an inside radius as small as one-half the diameter of the tubing by the use of the same technique, but with the addition of an internal fluid. By this method it is possible to fabricate shapes such as elbows, traps, nozzles, etc., where a permanent set is necessary.

Graphical Analysis of Periodic Loads

In the data sheet for graphical analysis of periodic loads by B. F. Raber and F. W. Hutchinson, published on pp. 241 and 242 of the April, 1942, HPAC, the first sentence in the discussion of Fig. 9 reads in part ". . . is valid for any wall of material for which $k/c\rho$ has the value for which this solution was obtained." While this condition is required, it is not sufficient and the sentence should be extended by adding "and for which the k/h ratios remain the same."

QUOTED . . . .

Coal Storage.—"I cannot tell individuals how much coal they should store, but as a general rule I think that a supply adequate for 60 to 90 days is not too much for the ordinary industrial consumer to have on hand," said Howard A. Gray, acting director of solid fuel coordination, at the annual meeting of the Chamber of Commerce of the United States. "Public utilities and vital war industries ought to carry coal enough to last at least 120 days. Householders should store their entire next winter's supply this summer, if that is possible. The coal in storage now generally does not even approach these figures."

SAVE .... SIMPLIFY .... SUBSTITUTE

SUMMARY—Reported here is the two-day "all industry conference of the air conditioning and refrigeration industry," which was held in Chicago last month. A feature of the affair was the "program for victory," a conservation plan endorsed by several industry organizations, which was presented by the president of the American Society of Refrigerating Engineers . . . Information on substitute materials, functions of various WPB branches, priorities, allocation of refrigerant, importance of service engineers and wholesalers, etc., was given by other speakers . . . The conference ended with an inspiring address on what the industry is doing to win the war by producing for victory, and the prediction that when peace comes after the inevitable defeat of the axis gang, war manufacturing experience will enable all the peoples of the world to achieve standards of living higher than ever before, and that consequently there is no reason to expect a severe deflation or depression

Conservation Plan Endorsed by Air Conditioning and Refrigeration Groups .... Substitute Materials, Priorities, Winning the War -- and the Peace -- Discussed at Chicago "All Industry" Conference

VOLUNTARY ACTION by individuals, industrial companies and industry organizations is imperative if this country is to overcome the enormous advantage gained by the axis powers through years of war preparation, Dr. William R. Hainsworth, president of the American Society of Refrigerating Engineers, declared in outlining a "program for victory" to the two-day "all industry conference of the air conditioning and refrigeration industry."

Dr. Hainsworth, vice-president in charge of research for Servel, Inc., said this conservation plan to "save, simplify and substitute" vital materials needed for war has been endorsed by 12 leading refrigeration and air conditioning organizations in the United States. The War Production Board, he said, has recognized the program as a "very desirable and important action on the part of the industry," and it has been endorsed officially by Donald Nelson.

The program calls on operating engineers, manufacturers, distributors and service firms to "avoid every waste of essential materials and supplies; to make the most of existing equipment; to collect reclaimable metal for wartime use; to release vital war metals by finding substitutes; and to cooperate in all possible ways with governmental agencies in the conservation of equipment, materials and power." Among other things, manufacturers are asked to pledge themselves to "research to find new applications to further the war effort."

Text of the program follows:

I pledge to conserve for victory—save . . . simplify . . . substitute . . .

If a refrigeration or air conditioning manufacturer, I will . . .

1) Use substitutes for critical materials. Eliminate waste and promote systematic maintenance.

2) Standardize equipment, test methods and practice. Simplify product line.

3) Urge users to make full use of service facilities to keep equipment in order.

4) Assist distributors' service organizations and jobbers with engineering and service help on this wartime program.

5) Seek out usable condensing units, fittings and other materials and use in place of new products.

6) Research to find new applications to further the war effort.

7) Prepare for air raids. Emphasize air raid instructions on refrigerating equipment.

If a distributor, dealer or jobber, I will . . .

1) Use my experience to conserve equipment and materials. Select proper sizes and types of equipment for most efficient use.

2) Speed up war production by refrigeration processes. There are many such applications. Discover new ones.

3) Check over installation and service facilities. Put my shop in order.

4) Employ capable, experienced and industrious service men. Encourage their training.

5) Urge users to have their refrigeration equipment checked and put in good order.

6) Take inventory of idle refrigeration equipment, valves, fittings and other materials. Use this equipment in new installations or broadcast its availability.

7) Arrange to give emergency service.

If an operator, maintenance or service engineer, I will . . .

1) Keep machines running efficiently and smoothly. Check performance. Conserve electricity, fuel and power.

2) Find all leaks and repair them as soon as possible. Be sure not to add re-

frigerant until leak is found and repaired. If necessary, remove refrigerant and transfer it to an approved container. Avoid purging to the air. Check shaft seal. Tighten gasket joints.

3) Check compressor oil level. Oil motor bearings and keep them oiled.

4) Keep motors operating efficiently. Clean commutators and brushes. Replace badly worn bearings.

5) Check belt alignment and tension. Conserve rubber.

6) Blow off condenser surface on air cooled machines. Be sure that air supply is adequate.

7) Clean condensers on water cooled machines. Be sure there is no partial stoppage. Be sure that the water valve is set to maintain economical condensing pressure. Clean regulating valve and strainers.

8) Check insulation for moisture, deterioration and tightness. Replace if defective.

9) Make certain that tools and equipment are in good order. Clean and check tool kit.

10) Check stock of parts, fittings and supplies.

11) Return empty refrigerant cylinders to suppliers without delay.

12) Give prompt service. Be sure of the diagnosis of trouble. Avoid guesswork. Plan work to save time and tires. Concentrate effort. Avoid repeat calls. Study service manuals and information supplied by manufacturers.

13) Check expansion valve if crankcase runs cold.

14) Remove oil from rubber mounting blocks.

15) Clean evaporator surface.

16) Advertise and publicize my ability and desire to aid in the wartime program.

17) Instruct users in procedure in an emergency.

Alternative Materials

In his talk on substitute materials and practices in the air conditioning and refrigeration field, George Allen, Mueller Brass Co., predicted increased use of steel pipe and threaded fittings even on war production projects with high priority ratings, because of the copper shortage. He stressed the point that with steel tubing, flares must be perfect because the material is less malleable than copper. He mentioned also that glass tubing was replacing stainless steel for dairy service, and that this was requiring changes in some local ordinances covering the cleaning of dairy service piping. Mr. Allen preferred the term "alternative material" to "substitute."

The critical position of our material supply was summarized in an excellent address by Harvey A. Anderson of the WPB bureau of industry conservation, who listed various strategic materials in the order of their scarcity. Paul Reed, Servel, Inc., pointed out that the refrigeration service engineer is of increased importance in wartime, as installations must be kept running despite difficulties in obtaining repair and replacement parts. Alex H. Holcombe, Jr., Victor Sales & Supply Co., reviewed the functions of the jobber and wholesaler, and said that the stocks of equipment, parts and tools carried in industrial areas give assurance that vital war production will not be interrupted. George Taubeneck, Business News Publishing Co., cited some of the war production advantages of air conditioning and refrigeration.

The functions of the air conditioning and commercial refrigeration branch of the War Production Board, and sidelights on its methods of operation, were given by Harry C. Williams, its assistant chief, assisted by George Meek, bureau of industrial conservation. In the question period, there was considerable discussion of the L-38 limitation order (in preparation at the time). Condensing units with wood bases, cast iron valves and steel flare nuts were predicted.

A question on the availability of "Freon" was referred to R. J. Thompson, Kinetic Chemicals, Inc. Mr. Thompson explained the method of allocating this refrigerant in accordance with the available supply each month, and said that there was now an ample supply. It is important, however, that the cylinders for refrigerants be returned promptly, as there is a cylinder shortage. It was also pointed out at this time that the Army and Navy Munitions Board had recently rescinded its April 1 order which had prohibited use of "Freon" in new installations having a capacity greater than 5 hp.

Henry A. Dinegar, of the WPB division of civilian supplies, spoke briefly on air conditioning and commercial refrigeration equipment in the civilian economy.

After the War

The last feature of the two-day session was a most inspiring yet practical address by A. B. Schellenberg, president, Alco Valve Co. He described the extreme importance of air conditioning and refrigeration equipment to wartime production and the contribution of the industry—which is accustomed to precision manufacturing—to the victory program. He stressed the point that, while the government is the principal buyer of air conditioning and refrigeration equipment, it is *not* "one customer". It comprises many, many customers throughout the country, numerous applications, and has every type of air conditioning and refrigeration problem. He indicated that the various divisions of the government must be served by the industry just as peacetime customers were served, but that because of the importance of winning the war, all of us must be more efficient and do everything each of us can to "keep 'em refrigerating".

In a straight from the shoulder and optimistic conclusion to his address, Mr. Schellenberg said that he saw no reason whatever for any deflation or depression after the successful conclusion of the war. World peace can be assured, he said, after the axis powers have gone down to their inevitable defeat, by raising the standard of living of all the peoples of the earth. As the living standard in China and India and all countries increases, so will that automatically of America. We will have enormously increased productive capacity, tremendously more experience with new materials and redesign of products, and will be better manufacturers because of the requirements which are now being met in producing war materials. Conversion of the automotive industry to war will prove to be a great advantage when peace comes, he predicted, for we will then have entirely new, better, cheaper and more automobiles. The great aircraft plants aren't going to shut down when the war ends, he said. Air cargo transportation will circle the globe; those who think we will travel by air and still ship our goods by slower methods just don't know the history of transportation, he remarked.

CODE FOR PRESSURE PIPING

SECTION 6—FABRICATION DETAILS

By Ludwig Skog, Chairman, Subcommittee on Fabrication Details; Partner, Sargent and Lundy

SUMMARY—Since the American Standards Association code for pressure piping (B31) was first issued as a tentative standard in 1935, there have been many developments and advances in piping practices, as described by Sabin Crocker in the January HPAC in the first of a group of articles, of which this is the sixth. The piping code has been thoroughly revised and is being reissued as a full American standard. Section 1 of the new code, on power piping, was reviewed by Alfred Iddles in the February HPAC. In March, the section on gas and air piping was covered by John S. Haug, and the section on oil piping systems was analyzed by A. D. Sanderson. The code sections on district heating piping, and on refrigeration piping (including air conditioning) were reviewed by G. K. Saurwein and A. B. Stickney in April. . . . Last month, part of the section on fabrication details was discussed by Ludwig Skog, who completes his review here.

NUMEROUS CHANGES in the art of welding which have developed since 1935 necessitated a complete revision of the welding rules in the chapter on welding pipe joints in section 6 of the new ASA code for pressure piping. Of particular significance are the new rules for procedure and operator qualification, recommendations for backing rings, rules for preheating, and new requirements for fillet welds to embrace the use of socket welding fittings.

Backing Rings—Where complete penetration is desired without projection of weld metal inside the joint, it is customary practice to use a backing ring for single U or V groove joints. While it was recognized that a number of designs of backing rings have proved acceptable, all of these could not be included in the code. Until such time as standard designs have been approved for backing rings, it was decided to include as recommended practice, four of the more widely used types as follows:

a) Where the nominal pipe wall thickness is ½ in. or less, or where it is not considered necessary or practical to machine the inside of the pipe, the recommended practice is as shown in Fig. 1.

b) Where the nominal pipe wall thickness is over ½ in. but is ¾ in. or less, and where it is considered necessary to machine the inside of the pipe, the recommended practice is as shown in Figs. 2 and 3. If the pipe wall is recessed for a backing ring, the depth of such recess shall be so limited that the remaining net section of the finished joint is not less than the minimum required thickness.

c) Where the nominal pipe wall thickness is over ¾ in., the recommended practice is as shown in Figs. 2, 3 and 4.

Welding Grooves—A subgroup of ASA sectional committee B16 on steel pipe flanges and fittings studied the details of welding grooves in general use. While it manifestly would be impossible to include every acceptable design of welding bevel, those shown in Figs. 5 and 6 were considered to represent the best known practice at the time, and were included as recommended practice in the American standard for steel pipe flanges and flanged fittings, ASA B16e. These designs are given the same status in the code for pressure piping.

Fillet Welds—The rapid development and use of socket welded fittings employed with small size pipe has necessitated careful consideration of fillet weld dimensions. A new American standard for socket welding fittings has been developed to meet the need for this product. Since the American standard calls for a design of fitting to be at least equal in strength to the pipe with which it is to be used, it is desirable, obviously, that the fillet weld which joins the two also be proportioned so that it is equal in strength to the component parts. Accordingly, dimensions shown in Fig. 7 were adopted. Numerous tests carried out by manufacturers and consumers have demonstrated that fillet welds made in accordance with these proportions will develop the full strength of the assembly.

The code requirement for fillet welds used for fittings other than flanges is noted as follows:

Fittings other than flanges may be attached by fillet welds, provided the pipe is inserted in the fitting and provided the fillet welds meet the requirements of Figs. 7 and 8.

Fillet welds also are permitted for the attachment of flanges of the slip on type providing the attachment is as shown in Fig. 8.

Preheating—No requirements concerning preheating were included in the last issue of the code, since its significance was not fully appreciated at that time. The new requirements call for preheating of welded joints in carbon steel having a carbon content in excess of 0.35 per cent with a nominal pipe wall thickness of ½ in. or greater, and in carbon molybdenum steel. Joints shall be preheated adjacent to the welding zone to a temperature of not less than 400 F and may be accomplished by any suitable method, provided the zone is uniformly heated and the temperature is maintained during the actual welding operation.

Stress Relieving—Requirements for stress relieving have been retained and expanded to cover carbon molybdenum as well as carbon steel. Welded joints in carbon steel shall be stress relieved when the nominal pipe wall thickness is ¾ in. or greater. Welded joints in carbon steel having a carbon content in excess of 0.35 per cent, and in carbon molybdenum steel, shall be stress relieved when the thickness is ½ in. or greater. Stress relieving is to be accomplished by heating uniformly to a temperature between 1100 and 1250 F, and held there one hour per inch of pipe wall thickness for carbon steel, and two hours per inch of thickness for carbon molybdenum steel, but in no case less than one-half hour.

Stress relieving may be accomplished by heating the complete structure as a unit, or heating a complete section containing the weld; or as is more common for pipe welding, heating a circumferential band containing the weld at the center. The width of the heated band is required to be at least equal to twice the width of the weld. The latter constitutes a distinct departure from the previous version of the code,

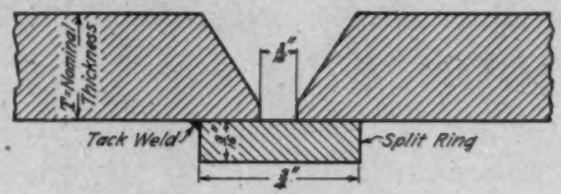


Fig. 1-V-butt joint with flat ring

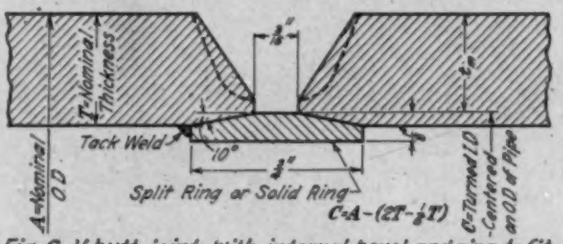


Fig. 2-V-butt joint with internal bevel and ring to fit

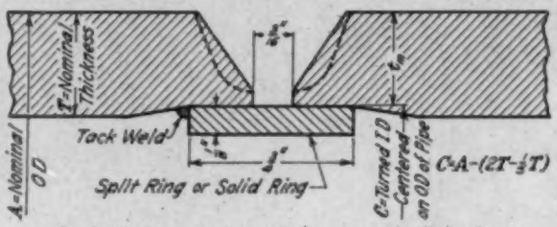


Fig. 3-Recessed V-butt joint with flat ring

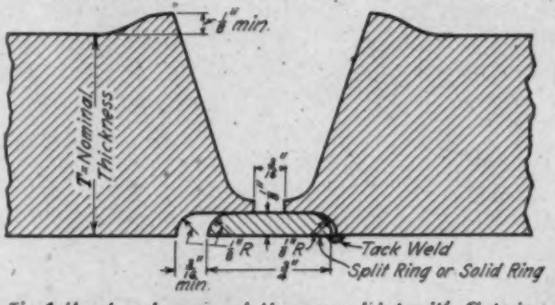
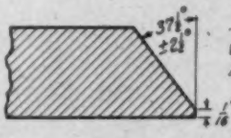


Fig. 4-Upset and recessed U-groove joint with flat ring



Standard straight bevel of welding ends for wall thicknesses up to 3/4 in. inclusive.

Fig. 5

Standard U bevel of welding ends for wall thickness greater than 3/4 in. This form of bevel is not recommended for acetylene welding and may be changed to straight bevel as in Fig. 5.

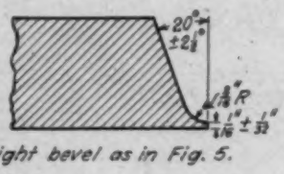


Fig. 6

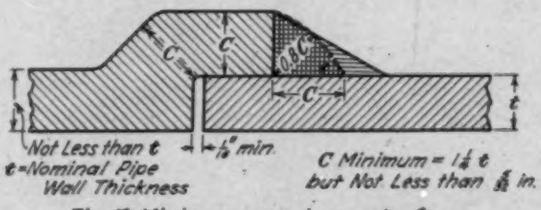


Fig. 7-Minimum requirements for socket and fillet weld dimensions

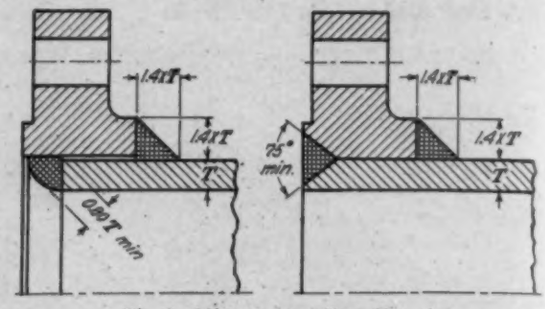


Fig. 8-Slip-on welding flanges

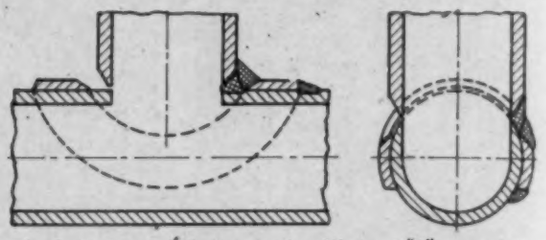


Fig. 9-Strap reinforcement with type "A" preparation

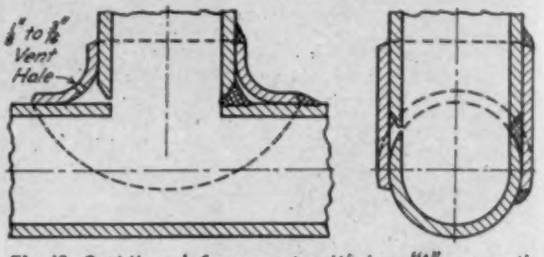
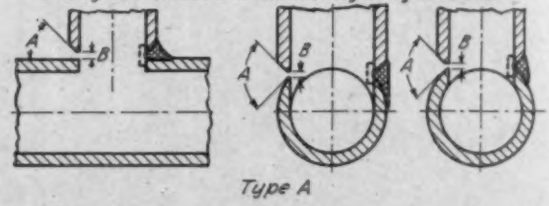
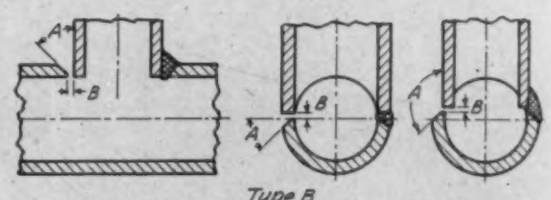


Fig. 10-Saddle reinforcement with type "A" preparation

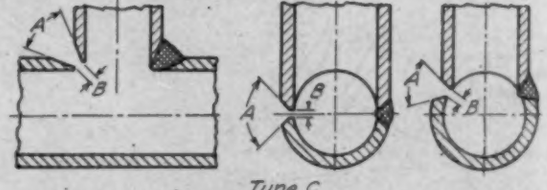
"B" May be Increased when Backing Ring is Used.



Type A



Type B



Type C

Angle "A" to be Not Less than 45° for any Wall Thickness of Pipe
"B" to be Not Less than 1/16" nor More than 1/4".

Fig. 11-Acceptable types of preparation for 90 deg full size and 90 deg reduced size branch connections

since the earlier edition required a width of band equal to six times the pipe wall thickness which, in case of the heavier pipe schedules, was impossible to attain for some pipe-to-valve joints. In the light of modern practice, the 1935 requirement is deemed unnecessary in any case.

No rules as to size of tack welds are specified in the revision of the code, but it is required that they shall be of the same quality and made by the same procedure as the completed weld. Otherwise, they shall be removed during the welding operation. In the larger sizes and heavier wall pipes, as many as four 2 in. tack welds are employed, which may remain as a part of the finished weld. Under these conditions, it obviously is essential that the tack welds should be of the same quality and made by the same procedure as the completed weld. This requirement does not prohibit tack welds made by gas welding where the finished weld is an electric weld, provided the tacks are completely melted out during the finish welding operation.

Operator and Procedure Qualification—Complete revision of the qualification tests for welding operators has been made. A new set of rules for qualification of the welding procedure has been added so that complete qualification for welding consists in first qualifying the procedure, and subsequently the operators thereunder. These rules are in agreement with the corresponding rules for qualification of operators and procedure formulated by the American Welding Society. Section IX of the ASME boiler code, recently issued, concerns qualification rules for welding operators and procedure which also are in agreement with the AWS rules. Thus, it is evident that progress is being made toward unifying the rules regarding welding, which heretofore have been most confusing due to the diversity between the several codes.

Procedure Qualification—In the procedure qualification rules, the types of test outlined are for the purpose of determining the tensile strength, elongation and degree of soundness of welded joints made under a given procedure specification. For welded butt joints, the type and number of tests are shown in the following table:

Maximum Thickness To Be Welded In Construction	Test Pipe Wall Thickness	Number and Type of Tests Required				
		Reduced Section Tensile	Free Bend	Root Bend	Face Bend	Side Bend
Up to and including ¾ in.	⅝ in.	2	2	2	2	
Over ¾ in.	Maximum but need not exceed 1 in.	2	2			4

Fillet weld soundness tests also are included. If fillet joints only are to be made, it is necessary, in addition to making the fillet weld soundness test, that the procedure shall be qualified for butt joints, but in such case only the reduced section tensile and the free bend tests for butt joints need be made.

If these tests are passed satisfactorily, the procedure is considered acceptable and should be recorded by the manufacturer or contractor as a "procedure specification." Acceptable types of forms for so doing are given in appendix II of this section of the code. If any changes are made in the procedure, the procedure specification must be revised or amended to show the changes. If the change is one of those listed in paragraph 1(b) of appendix I, it may be necessary to requalify the procedure or the operator, or perhaps both, as specified therein.

Operator Qualification—The qualification tests specified for operators are specially devised to determine the operator's ability to produce sound welds when using a previously qualified procedure. The tests used for operator qualification are as follows:

a) Welded butt joints—(1) root bend test; (2) face bend test; (3) side bend test.

b) Fillet welded joints—(1) Fillet weld soundness test.

The number and type of tests required for butt joints in pipe are shown in the following table:

Thickness for Which Operator Is To Be Qualified	Thickness of Material for Test Weld	No. and Type of Tests Required		
		Root Bend	Face Bend	Side Bend
Up to and including ¾ in.	⅝ in.	2	2	
Over ¾ in.	Maximum but need not exceed 1 in.			4

It may be observed that a significant difference in these rules is that the only test required for operator qualification is some form of bend test—either root, face or side bend. The previous version of the welding rules did not distinguish between procedure and operator qualification, and the reduced section tensile and nick break tests were important features of the former operator qualifications. The sole criterion of the operator's qualification to make a satisfactory welded joint is his ability to produce a sound weld as judged by the above tests.

The welding operator, having successfully passed the qualification tests, need not be requalified unless: (1) the welding operator is not engaged in a given process of welding (i.e, arc or gas) for a period of three months or more; or (2) there is some specific reason to question the operator's ability.

No mention is made here of the numerous detail requirements of the procedure and operator qualification rules. The complete rules may be found in section 6, appendix I, parts I and II, of the code.

Branches and Specials

Welded Branch Connections and Fabricated or Cast Specials—This chapter, representing an entirely new addition to the section on fabrication details of the code, concerns the methods to be followed in the fabrication of intersecting welded branch connections, and is limited principally to those branch connections in which the angle between the outlet and the run is not less than 85 deg. All of the requirements of the chapter on welding, which immediately precedes this chapter, are to be followed insofar as they apply.

Development of Rules—It has been recognized for some time that a branch connection welded into a straight length of pipe reduces the

ability of the assembly to withstand pressure, so that yielding under internal pressure is reached at a considerably lower value than for a straight length of pipe. Since the shapes are too complicated to permit accurate theoretical analysis, it has been necessary to rely on tests of full size manifolds to deduce the rules for reinforcement of branch outlets so that strength equivalent to that of the straight pipe can be secured. A special subgroup on welded branch connections of the sectional committee was organized to study the various types of reinforcement and welding grooves with the object of formulating rules suitable for inclusion in the code for pressure piping. Consumers and manufacturers participated in the test program, and the results were carefully analyzed. Important features of the rules which were formulated are:

a) A header having a branch or branches attached by welding shall be reinforced to make it equivalent in strength to the computed bursting strength of straight sections of pipe as determined by the formulas for pipe wall thickness given in sections 1 to 5 of this code.

The necessity for and the amount of reinforcement required in the crotch section shall be determined by the rule that the area available for reinforcement shall be equal to or greater than the required area as defined as follows: The required area shall be determined as the cross sectional area of the finished opening in the header, considered as a rectangle whose length is the diameter of the finished opening and whose height is the theoretical minimum wall thickness of the header. The hole to be compensated for by reinforcement shall be measured on the diameter of the outlet parallel to the axis of the run. The available area included as reinforcement shall be at least the sum of the following cross sectional areas—

(1) The cross sectional area of the fillet welds adjacent to the branch and above the outside surface of the header.

(2) The cross section area of the branch determined from the minimum thickness on inspection and a height equal to 2½ times the minimum thickness of the branch wall above the outside surface of the header, or above the outside of the reinforcing ring or saddle, if used.

(3) Any additional thickness of the wall of the header above the theoretical minimum thickness required by the wall thickness formulas of sections 1 to 5 incl. for a distance equal to the inside diameter of the branch on each side of the center line of the branch.

In case the sum of these available areas is less than the required area, additional reinforcement shall be provided. This additional reinforcement shall be obtained by additional thickness of header wall and/or branch wall, ring, saddle and/or weld metal. See Figs. 9 and 10.

The material used for ring or saddle reinforcement may be of specifications differing from those of the pipe, provided the cross sectional area is made in proportion to the relative strength of the reinforcement and pipe materials at the operating temperature and provided it has welding qualities comparable to those of the pipe. No credit shall be taken for the additional strength of material having a higher strength than that of the part to be reinforced.

When rings or saddles are used which cover the weld between branch and header, a vent hole shall be provided in the ring or saddle to reveal leakage in the weld between branch and header and to provide venting during welding and heat treating operations.

The use of ribs or gussets shall not be considered as contributing to reinforcement of the branch connection. This does not prohibit the use of ribs or gussets for purposes other than reinforcement.

Fillet welds used in attaching the reinforcement shall be full fillet welds.

Reinforcing rings and saddles shall be accurately fitted to the parts to which they are attached. Figs. 9 and 10 illustrate some acceptable forms of reinforcement.

b) Branch connections attached at any angle less than 85 deg to the run become progressively weaker as the inclination becomes less. Any such designs must be given individual study and sufficient reinforcement must be provided to compensate for the inherent weakness of such construction. The use of encircling ribs to support the flat or reentering surfaces is permissible, and may be included in the strength calculations.

Types of Reinforcement and Preparation—The two acceptable types of reinforcement mentioned in these rules are the strap or ring types shown in Fig. 9, and the saddle type shown in Fig. 10, both of which are widely used.

Acceptable types of preparation for welded branch connections which have been proved to be satisfactory by tests are shown in Fig. 11. Type *A* represents the practice in which the opening in the header is cut straight and the branch sets on top of the header. Type *B* represents the practice whereby the opening in the header is scarfed around its entire circumference but the branch is cut square. Type *C* represents the practice where both the header and branch are scarfed. In all these types of preparation, it has been demonstrated that it is possible to secure acceptable fusion and to produce sound welds.

Specials—While the preceding requirements pertain to welded branch connections, rules for special fittings fabricated by welding, and for cast special fittings, are included in this chapter also. These are as follows:

a) Fabricated swages, orange peel bull plugs, segmental elbows and other special fittings may be formed from pipe or plate material and welded in accordance with the general requirements for welded joints specified in this section. Design shall be such that all special fittings will be equivalent in strength to the computed bursting strength of a straight section of pipe as determined for the service conditions by the formulae for pipe wall thickness given in sections 1 to 5 inclusive in this code. The inherent weakness of such designs shall be taken into account in establishing the service conditions under which they may be used.

b) Cast special fittings with welded ends shall conform to material requirements of ASTM specifications A216 or A217, and shall be designed in accordance with requirements of American standard B16e, or paragraph 634 if practicable. Where rules of American standard B16e are applied in the design of such fittings, the shape factor employed in determining minimum wall thickness may be omitted for the cylindrical or tapered ends. When welding ends of castings are finish machined both inside and outside and carefully inspected, the quality factor at the welding ends may be taken as 100 per cent, but in no case shall the thickness of the welding end be less than the wall thickness of the mating pipe.

It should be remembered that all requirements given in the code for pressure piping are minimum requirements deemed essential for safety, and any construction superior from a safety viewpoint is acceptable. These code rules on reinforcement for welded branch connections are quite similar to those prescribed in the ASME boiler construction code, except that the latter are not intended to be applied to branch openings greater than 60 per cent of the shell diameter.

The subgroup does not submit that these rules represent the last word on the subject, but they are considered to comprise a workable set of rules which will give satisfactory results for the ordinary classes of service covered by this code. Further consideration and modification of the rules, if necessary, will be a subject for future consideration of the subgroup.

INDUSTRIAL HEALTH DEFENSE

Dust Counts and Their Significance

By N. R. Bernz

SUMMARY—Considerable attention has been focused in recent years on the conservation of health in industry, a subject which becomes of increased importance in connection with the production for victory program. Mr. Bernz, of the Fidelity and Casualty Co. of New York, discusses the methods employed in the evaluation of dust exposures, reviewing the types of sampling and counting instruments and the various factors involved. . . . In March he covered the Greenburg-Smith impinger—best known of the sampling devices—and the Sedgwick-Rafter and Hatch cells for dust counting. This month, he discusses the midget impinger, the microprojector, the Owens jet, the B. & L. dust counter and the konimeter. . . . In his next article he will consider the choice of dust counting instruments, methods of sampling and evaluation. . . . His material is based on a paper read before the Greater New York Safety Congress

THE DESIGN of the midget impinger is—as the name implies—a duplication of the impinger described in the March HPAC, except that it is of much smaller proportions. The volume of the sampling liquid is usually confined to 10 cc and the sampling rate to 1/10 cfm, instead of 75 cc and 1 cfm, respectively, for the larger instrument. [See Bureau of Mines Midget Impinger, by H. H. Schrenk and Florence L. Feicht, U. S. Bureau of Mines, I. C. 7076, June, 1939]. The method of evaluation of the dust samples is the same for the two instruments.

Referring to the equation on p. 175 of the March HPAC, and assuming a 10 min sample, the factor which in the case of the larger instrument was 20,000 now becomes 40,000; this is due to a higher ratio between the volumes of sampling fluid and air. This will require less dilution, however, so that the factor influencing the error remains the same. A 10 million particle sample as referred to in the March issue would mean 250 particles in the field. This would require a dilution of only five times, and the final factor would again be 200,000.

The Microprojector — Recently, the microprojection method has been used for quantitative determination of impinger samples. [See A Microprojection Method for Counting Impinger Dust Samples, by C. E. Brown, L. A. H. Baum, W. P. Yant and H. H. Schrenk, U. S. Bureau of Mines, R. I. 3373, 1938]. The difference between this method and the regular microscopic method is that the microscopic field is projected on a translucent screen and at a much greater magnification. Through a combination of eye piece, objective, tube length and distance of the screen, a magnification of 1000 diameters is obtained, as compared to about 100 times used in the standard method described before.

The screen is usually ruled into large squares and subdivisions in such a manner that the largest squares represent 1/100 sq mm each of the microscopic field. With a 1 mm deep cell they also represent 1/100 cu mm each of the dust sampling liquid. Brown *et al* indicated that dust counts made with this technique result in as great precision as counting with the direct microscopic method if a field of five squares is counted. On this basis, 1 cc of sampling liquid corresponds to 20,000 fields, so that the number of particles per cu ft of air will be:

$$\frac{\text{Count per "Five Square" Field} \times 20{,}000 \times \text{cc}}{\text{Cu Ft of Air}}$$

If the count per field is called C and we assume a 10 min sample with the midget impinger (i.e., 1 cu ft of air and 10 cc sampling liquid) the above takes the following form:

$$\frac{C \times 20{,}000 \times 10}{1} = C \times 200{,}000$$

Although a smaller field has been covered in the actual counting, the above factor is no greater than those developed for the direct microscopic method on impinger samples. This is due to the fact that with this tech-

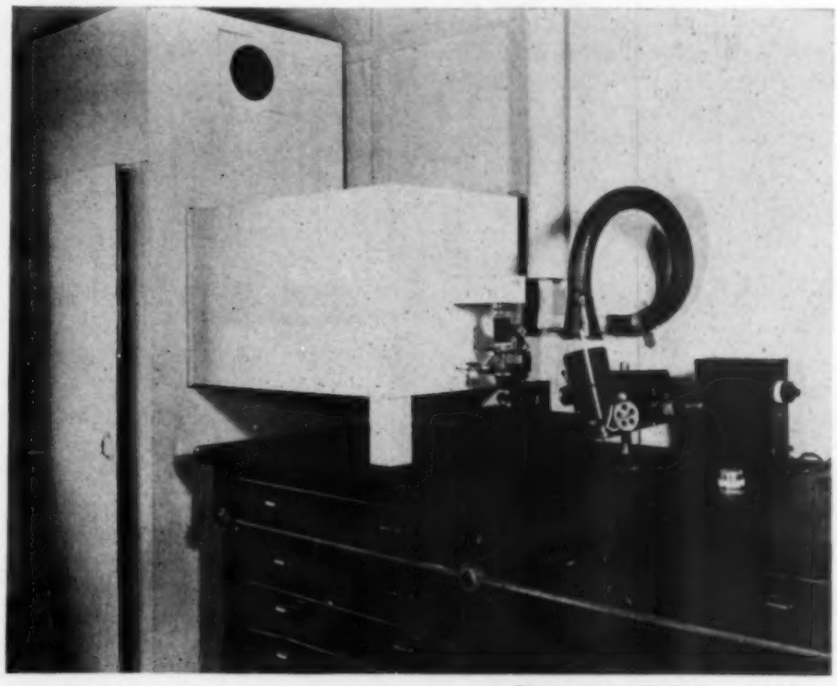


Photo courtesy U. S. Bureau of Mines.

The microprojector is one of the methods described this month

nique no dilution needs to be considered until exceedingly heavy concentrations are encountered. According to Brown and his colleagues, 50 times as concentrated samples can readily be counted with this method as those permissible in the regular microscopic procedure. For a 10 million particle sample, a field count of 50 is required. This corresponds to a concentration of 1000 particles per cu mm of the sampling liquid, as compared to about 200 recommended for direct microscopic observation, which is only five times. Based on the above-mentioned claim, with this technique it should be possible to count dust concentrations up to 100 million particles per cu ft of air, without dilution, i.e., a calibration factor not over 200,000.

In view of the heavy concentrations that can be counted by this method, large air volume samples taken in moderately dusty atmospheres can still be counted without dilution, thus reducing the multiplication factor. If, for instance, the standard impinger with 75 cc of sampling liquid were operated for 30 min in an atmosphere with 10 million particles per cu ft, the factor would be reduced to 50,000 which is in the class with that of the konimeter, and the dust concentration of the sampling liquid would still be only 20 times that of the regular technique. This factor, however, would not cover a concentration range exceeding 25 million particles per cu ft. [The above discussion is based on a field of 0.05 cu mm.] Obviously a heavier concentration can be counted with a low factor if the size of the field is increased, and for a certain concentration a large factor similarly can be decreased.

It is apparent that practically the same result will be obtained by the microprojection technique, if the final calculation is based on one square instead of five, provided that the larger area has been covered in the actual counting and the value thus obtained reduced to an average per square to which a larger factor is applied. On this basis, the factor will be five times higher, or 10^6 in the case of the midget impinger sample referred to above. Excluding the deviation produced by multiplication of a slight mathematical error made in obtaining the average count per square, which is insignificant when compared to errors produced in the counting itself, this method is obviously just as accurate.

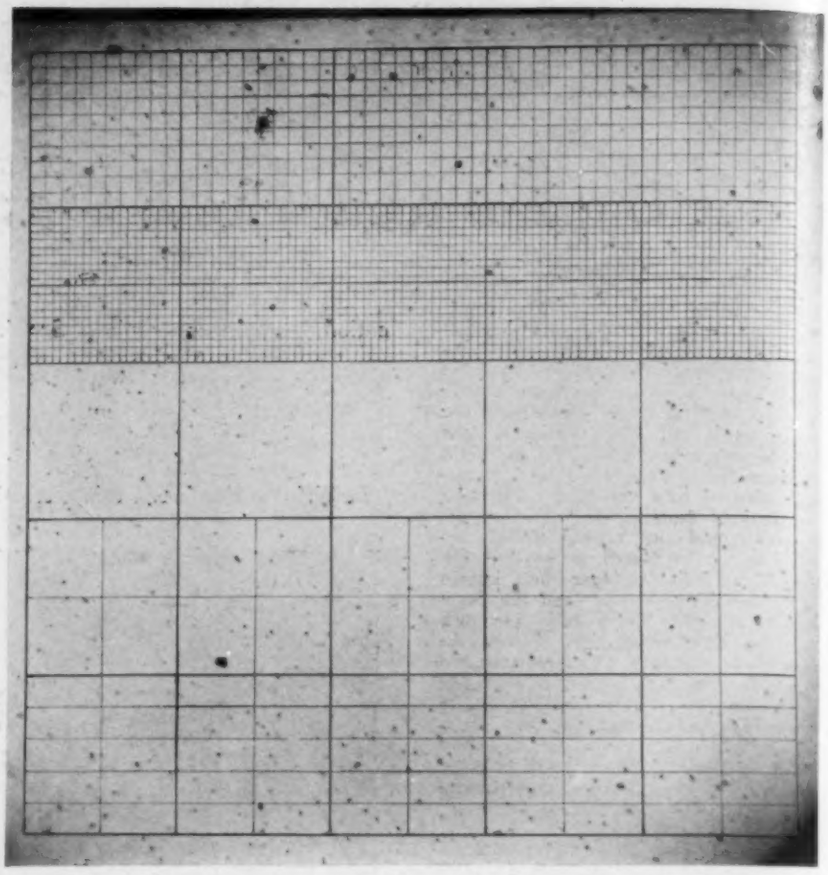


Photo courtesy U. S. Bureau of Mines.

Microprojector screen, showing shadows of dust particles

On the other hand, if the actual enumeration should be confined to such a small liquid sample as represented by one square, the final error might be considerable after applying a factor of one million.

Owens Jet — This instrument consists of an air-entrance tube open at one end and communicating with a small airtight chamber at the other end through a 10 mm by 1 mm streamlined rectangular nozzle. [See Industrial Dust, by Philip Drinker and Theodore Hatch, McGraw-Hill Book Co.]

An impinging surface is provided by means of a circular glass slide placed opposite the nozzle in the chamber referred to. To this chamber is connected a 50 cc capacity hand pump which is the means of suction. To operate this instrument the chamber is brought under negative pressure by means of the hand pump; the dust laden air rushes in through the entrance tube and the nozzle, into the airtight chamber where it impinges against the slide.

Owens jet dust samples are evaluated in the following manner: The glass slide containing the dust is placed under a high power microscope and counts made by means of light field illumination. The microscopic setup involves besides the light field condenser, a 2 mm oil immersion objective and a 15 x eye piece containing an ocular micrometer ruled into squares. This combination gives a magnification of over 1000 diameters. Considering the shape of the nozzle, it is evident that the dust record on the glass will appear in the form of a band approximately 10 mm long by 1 mm wide. When counting these samples it is practice to count several ribbons of the same width as a certain square in the eye piece, across the dust record, and to obtain an average count of such ribbons. For calculation of the final count the number of particles in such a ribbon is multiplied by the length of the dust record in units of ocular micrometer squares and divided by the number of cc's of air volume, which is 50 per pump stroke for this particular instrument. This will give the dust concentration in terms of particles

per cubic centimeter of air. If the average number of particles per ribbon is called C and only one stroke is made, the equation for particles per cc of air can be written:

$$\frac{C \times \text{No. of Ocular Micrometer Squares across the ribbon}}{50}$$

In order to see how the calibration factor may affect the final count in this case, let us assume an ocular micrometer ruling which at the particular magnification will make the sides of the squares 20 microns. Therefore, the length of the dust record which is 10 mm long will be 500 in terms of ocular micrometer squares. The number of particles per cc of air will therefore be:

$$C \times \frac{500}{50} = C \times 10$$

(Based on one stroke)

The count per cc of air would therefore be obtained by multiplying the average ribbon count by 10. In order to get the count into numbers per cubic foot of air, which is standard practice in this country, this value will have to be multiplied by $28.3 \times 1000 = 2.83 \times 10^4$. In other words, the count per cubic foot of air becomes $C \times 2.83 \times 10^5$. The final factor, therefore, influencing the original count, is in the same class as those developed for some of the methods previously described. While it is true that this factor will be brought down as the number of strokes is increased, the ribbon concentration will increase accordingly with the possibility of a greater error in ribbon count.

The Bausch & Lomb Dust Counter—This instrument operates on the same principle as the Owens jet and a number of the design features are common to both instruments; for example, the air entrance chamber, nozzle, circular glass slide and hand pump. The nozzle in the standard instrument is but 6 mm long by 0.4 mm wide and the hand pump has a capacity of only 28.3 cc. The principal difference, however, between the two devices is the microscope combined with the sampling mechanism in the B. & L. instrument. The microscope is arranged for a modified dark field illumination, i.e. an Abbé condensor with a central stop, 200 times magnification (10 x objective and a 20 x eye piece) and a micrometer disk in the ocular on which a grid is ruled in the form of a ladder 60 microns wide.

The dust record impinged on the circular glass slide in this instrument as in the Owens jet takes the shape of a band. The counting is also done in a similar manner, i.e. several ribbons 60 microns wide are counted across the dust record and an average of such counts is used in the final calculation. Since the length of the dust record is 100 times the width of the ribbon counted (6000 and 60 microns respectively), the count must be multiplied by 100 in order to include all the particles in the sample, and since one stroke represents only 1/1000 of a cubic foot of air, the above should also be multiplied by 1000 to get it on a cubic foot basis. In other words, the original count must be multiplied by 100,000 to obtain the final dust value per cubic foot of air. If the average count across the dust record is called C, the number of particles per cubic foot of air based on one stroke will be $C \times 100{,}000$.

The reason the multiplication factor becomes so much smaller in the quantification of samples by this method as compared to the Owens jet samples is simply that due to the microscopic setup, a much wider ribbon is counted across the dust record in the B. & L. instrument. In view of the greater magnification and narrower ribbon observed on the Owens jet samples, the counting error is no doubt smaller, so that although a larger multiplication factor is applied it does not follow that the final error is any greater.

The Konimeter—Basically the sampling mechanism of this instrument closely resembles that of the Owens jet and the B. & L. sampling devices. It is provided with an entrance nozzle, which in this case is of a circular shape, 0.6 mm diameter. The air is actuated by means of a pump and impinges upon a round glass plate as in the other two devices referred to above. In the modern instrument this glass plate is large enough and can be rotated so that 30 dust samples (spots) may be collected on a single disk. The average sampling volume is said to be about 5 cc of air per stroke. In this instrument, as in the B. & L. counter, a microscope has been incorporated for direct counting. The microscopic setup provides for dark field illumination, 200 diameter magnification and a grid in the ocular ruled in such a manner that two lines cross at an angle of 9 deg.

In the counting procedure, the point of intersection of these two lines is centered over the dust spot so that the sectors formed by the crossed lines divide the dust spot symmetrically.

For enumeration of the particles, it is practice to count all the particles in one 9 deg sector and then rotate the eye piece until four such sectors have been counted. For the final evaluation, the number of particles in four 9 deg sectors is multiplied by 10 and divided by the number of cc of the air sample, which gives the particles per cc.

If we call the average count per four 9 deg sectors C and assume a one stroke sample, i.e., 5 cc, the number of particles per cc can be stated as $(C \times 10)/5 = C \times 2$.

In order to transform this into particles per cubic foot of air, it is necessary to multiply by 28,300, which, therefore, $= C \times 56{,}600$.

This is a small factor compared to those developed for some of the other methods, and will naturally have a proportionally smaller effect on the actual dust counting error. Before we conclude, however, that a more exact dust count is obtained on a sample collected by this instrument, let us see what the proportion of the original error might be.

According to Mavrogordato [see Industrial Dust] anything up to 500 particles per cc is "easy to count."

This concentration corresponds to 14.2×10^6 particles per cu ft. While it is true that the term "easy" is relative, one would be inclined to believe it implies that the samples can convenienty be counted with a reasonable percentage of error. On this premise and considering the low multiplication factor, this instrument should probably be considered fairly accurate for evaluation of its own samples in concentrations below the 14 million class. Above this figure, Mavrogordato's classification indicates that counting is "not easy." This probably means that at this concentration the particles begin to pile on top of each other and as soon as that happens the error in counting no doubt becomes considerable. Therefore, although the multiplication factor is small, the final error may be just as great as with any other method when the above concentration class is exceeded.

Humidity Control in UNDERGROUND BOMBPROOF SPACES

By Edward Ledoux

SUMMARY—The author was until last year chief engineer of a French firm engaged in air conditioning engineering, particularly with adsorbents. Although many varied problems for the French army and navy were handled and solved, most of the firm's activity after 1931 was concerned with bombproof shelters, storerooms, forts and factories. . . . The necessity for humidity control in underground spaces was imposed by the considerable deterioration of equipment or supplies and numerous cases of pneumonia among men manning forts or servicing storerooms; dormitories in forts could not always be occupied steadily as they were meant to be. At first, no one had any idea either of the order of magnitude of the capacities required or of the most adequate means; wall evaporation was not suspected because the walls looked dry. . . . The bulk of this discussion bears on wall evaporation. This is intentional because it has been the author's experience that this source of humidity is always underestimated, if not disregarded entirely. He stresses the importance of this factor and how to cope with it; other sources of humidity are merely mentioned because their handling is just usual air conditioning. . . . In the January HPAC, Mr. Ledoux stressed the importance of the wall evaporation moisture factor, and enumerated means of reducing the conditioning load. A third article, to appear later, will cover isothermal and intermittent operation and the case of factories and shelters

A SHORT discussion of the mechanism of wall evaporation in underground spaces, and numerical examples, will serve to illustrate the author's comments [on pp. 6-8 of the January HPAC] on the importance of this factor in air conditioning of underground bombproof spaces. We will first consider the case where the arrival of capillary water at the wall surface is equal to the removal by evaporation and for the moment, we will assume complete abstraction of the original air present in the spaces when conditioning starts; as will be seen later, this original air is theoretically evacuated in the first few hours.

Wet Bulb Temperature

In the case of storage spaces where no heat is evolved, it is only in the particular case where the wet bulb temperature of the air is equal to the natural temperature* of the unconditioned space that the humidification line is one and the same as the adiabatic humidification line on the psychrometric chart. Generally, the humidification line varies continuously with time due to the change in wall surface temperature caused by heat exchange between air and masonry; this is illustrated on Fig. 1, where the natural initial condition of the space is 50 F saturated. If the point representing the conditioned air supply is O, the humidification line is OC at all times; if it is B, with a higher wet bulb temperature, the humidification line varies from BC at "time zero" to BD at "time infinite" and the wall temperature rises continuously. If the point representing the conditioned air supply is A, the humidification line varies continuously from AC to AE and the walls are cooled. In both cases, the humidification line tends toward the wet bulb line and operation tends to become adiabatic.

Effectively, choosing as main variables the total heat Q in Btu per lb of dry air and the absolute humidity w in lb of moisture per lb of dry air, and taking for origin of coordinates the natural initial conditions at the wall surface, the general equation of the humidification lines, in the present case of low vapor pressures is:

$$\frac{Q}{Q_o} = \left[\frac{Q - kw}{Q_o - kw_o}\right]^{\frac{m}{k+m}} \quad \ldots\ldots [1]$$

We also have:

$$\frac{Q}{Q_o} = e^{-\frac{qs}{Ak'(k+m)X}} \quad \ldots\ldots [1']$$

In these relations, which will permit plotting, Q_o and w_o are the conditions of the inlet air; k is the slope of the saturated vapor curve plotted as total heat versus absolute humidity and, in the range considered (50 to 60 F) it may be considered constant without great error and equal to 1700 Btu per lb of dry air for 1 lb moisture increase per lb of dry air; l is the distance in ft from the inlet grille; s is the wall surface per ft of air travel, in sq ft; and A is the flow in lb of dry air per hr. The value of m is:

$$m = \frac{qa}{HBk'X}$$

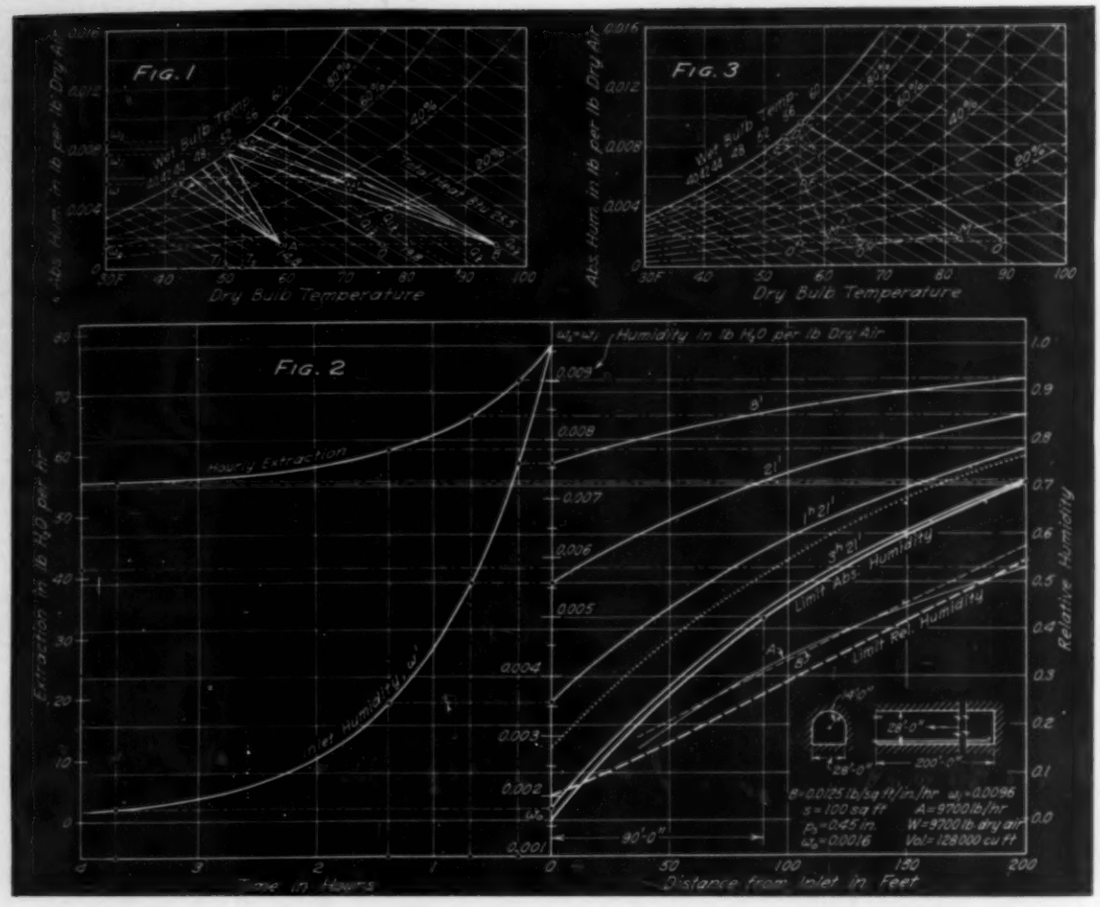
in which q is the conductivity of the masonry in Btu per sq ft per F for 1 ft thickness; a is the constant ratio of densities between air and water vapor ($a = 0.623$); H is atmospheric pressure in inches of mercury; B is the coefficient of evaporation in lb per hr per sq ft for 1 in. vapor pressure difference; and k' is the slope of the saturated vapor curve plotted as humidity versus temperature ($k' = 0.00034$ lb. per lb for 1 F temperature increase). X, which depends upon time t and diffusivity only, is the horizontal projection of the initial tangents to the temperature-space curves inside the walls. For X smaller than about 30 per cent of the wall thickness:

$$X = \sqrt{\frac{2q}{cd}t}$$

in which c and d are the specific heat and density of the masonry. Beyond the mentioned limit, the temperature of the outer wall surface begins to rise appreciably and X will increase more slowly due to heat loss to the earth whose diffusivity is greater than that of the walls. For thick walls such as those encountered in bombproof construction, however, the error committed by extended use of the above relation will only be of the order of 1 or 2 per cent. The initial tangents can also be determined from a Schmidt† graph although, taking the first segment for the tangent will give higher values than those corresponding to the true tangent. Although the precision of computation increases with the number of divisions used the error on the projection of the tangent is most likely to exceed that committed when extending the use of the previously given relation.

*The "natural temperature" is the temperature of the spaces and walls before any conditioning (50 to 60 F in temperate zones). See January, 1942, HPAC.

†E. Schmidt: A. Föpples Festschrift. J. Springer. Berlin. 1924. Also: W. Trinks. Industrial Furnaces, Vol. 1, John Wiley & Sons, Inc., 1934.



We will forego the calculations establishing the equation of the humidification lines because, although they are not particularly complicated, they are somewhat lengthy.

It will be seen however that for $t = 0$, $X = 0$ and m is infinite, so that the expression reduces to $Q/w = Q_o/w_o$, which is the equation of a straight line joining the air inlet conditions to the initial conditions. For t infinite, X is infinite and $m = 0$; the expression reduces to $Q = Q_o$, which is the adiabatic humidification line from the point representing the inlet conditions. For intermediate values of time, the lines are curved and concave toward the initial conditions at the wall surface. The temperature of the wall surface, which varies not only with time but with the location, is given by the intersection of the tangent tt' to the humidification line with the saturated vapor curve (Fig.1). Of course, for inlet conditions at O, the surface temperature and absolute humidity remain equal throughout to the initial temperature and humidity of the spaces and $w_o = w_1$ while $Q = Q_o = 0$.

Adiabatic Operation

Generally speaking, it will be found impossible or prohibitive to design the conditioning equipment for delivery of air at a lower wet bulb temperature than the natural temperature of the space. This is due to the fact that both temperature and dew point must be low while little or no cooling water is usually available. The best that can be done is to produce air at a wet bulb temperature equal to the natural temperature of the spaces. Operation is then adiabatic.

Let us determine the absolute humidity in the spaces under adiabatic conditions.

Stating that the humidity increase of the flow of air as it travels across the surface dS of wall area is equal to the evaporation from that surface, we have:

$$Adw = B\ (p_1 - p)\ dS$$

in which A is the flow of air in lb per hr, p is the vapor pressure of the air and p_1 the vapor pressure at the surface.

Considering that $\frac{w}{a} = \frac{p}{H - p}$, the expression may be written in terms of absolute humidity:

$$Adw = \frac{B\ (H - p_1)}{a + w}\ (w_1 - w)\ dS$$

If s is the wall area per foot travelled by the air, then $dS = sdl$, in which l is the distance from the inlet grille.

The vapor exchange equation may be written

$$\frac{Bs\ (H - p_1)}{A}\ dl = \frac{a + w}{w_1 - w}\ dw \quad \ldots\ldots\ldots [2]$$

which after integration becomes

$$\frac{Bs\ (H - p_1)}{A}\ l = -\ (a + w_1) \log \frac{w_1 - w}{w_1 - w_o} - (w - w_o)$$

Efficiency is the ratio between results actually achieved and the maximum possible. Expressed in terms of absolute humidity increase, the efficiency of humidification of the flow of air is therefore:

$$E = \frac{w - w_o}{w_1 - w_o}$$

since w_1 is the absolute humidity at the wall surface.

In the light of this expression, the last given equation may be written

$$\frac{Bs\,(H-p_1)}{A}\,t = -(a+w_1)\log(1-E) - E\,(w_1-w_0) \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

Plotting is easier under this latter form by dividing (w_1-w_0) into 10 equal intervals. The last item, $E(w_1-w_0)$, is very small and could be neglected without great error; so doing yields an approximate expression of the absolute humidity efficiency which will apply for low vapor pressures as in the present case:

$$E = 1-e^{-\frac{Bs\,(H-p_1)}{A\,(a+w)}\,t}$$

or, considering that $\frac{w_1}{a} = \frac{p_1}{H-p_1}$:

$$E = 1-e^{-\frac{Bs\,(H-p_1)^2}{aAH}\,t} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [4]$$

Neglecting air pressure variations entirely, as can be done in the present case of low vapor pressures, the efficiency is:

$$E = 1-e^{-\frac{BsH}{aA}\,t}$$

A numerical application of equation 3 is given on Fig. 2, the data being that indicated in the lower right hand corner; the natural temperature of the spaces considered is 56F and the conditions of the inlet air are represented by point O (Fig. 3). Operation is in a closed circuit. The result is the heavy line curve marked "Limit Absolute Humidity." Once this curve is plotted, the relative humidity can be read off the psychrometric chart along the 56F wet bulb line; the relative humidity curve is plotted in dash line and marked "Limit Relative Humidity."

Equation 3 does not take into account the original air present in the spaces when conditioning starts so that the curve shows the humidity in the spaces once the original air has been completely evacuated; up to that time, which is theoretically infinite, the humidity in the spaces is higher. Let us determine the rate of elimination of the original air and the decrease in humidity, making abstraction this time of wall evaporation.

Expressing that the decrease in humidity in the spaces is equal to the amount taken up by the flow of air, gives:

$$A\,(w'-w_0) = -W\frac{dw'}{dt}$$

$$\text{or } -\frac{A}{W}\,dt = \frac{dw'}{w'-w_0}$$

W being the weight of dry air originally present in the spaces and w' the outlet humidity of the air flow. Integrating the above yields:

$$-\frac{A}{W}\,t = \log\frac{w'-w_0}{w_1-w_0}$$

$$\text{or } \log\frac{1}{1-E} = \frac{A}{W}\,t$$

In the present numerical example we have one air change per hour $(A/W = 1)$ and the evacuation curve is plotted in the left hand half of Fig. 2 (curve marked "Inlet Humidity"). It will be noted that, although the time required for complete evacuation is infinite, for all practical purposes it can be considered finished in the first four hours.

To take care of the evacuation of the original humidity, w_0 must be replaced by w' in equation 3 which means considering the inlet conditions as w'; this has been done on Fig. 2 for the values of w' after 8′, 21′, 1ʰ21′ and 3ʰ21′ as shown by the dot and dash witness lines which determine the inlet humidity and the origin of the humidity curve at the considered time; the plotting of the rest of the curve is done by means of equation 3 and dividing the interval $(w_1 - w')$ into tenths.

Capacity

The hourly water vapor extraction from the spaces which must be handled by the conditioning equipment is plotted in the left hand half of Fig. 2. It is the product of the difference between inlet and outlet humidities and the flow. Due to the decrease in outlet humidity during the evacuation period, the capacity decreases continuously until a practical equilibrium is reached (55 lb per hr in the present case) when the entire capacity is spent against wall evaporation.

Natural Temperature

For low vapor pressures, the variations of the air pressure $(H - p_1)$ with p_1 are very small and have little effect on the expression of the efficiency given by equation 4. In other words, the efficiency remains substantially the same for all wet bulb temperatures within the range of underground conditioning. For this reason, a low natural temperature is a favorable factor since, the lower the absolute humidity, the smaller the plant capacity required.

This is illustrated on Fig. 2 by the curves A and B which respectively represent the limit absolute and relative humidities for a natural temperature of 50F instead of 56, all other conditions remaining the same. The plant capacity has dropped from 55 lb per hr to 41 lb per hr at the cost of only a slight rise in relative humidity.

Any circumstance which increases the humidification efficiency is an unfavorable factor. Equation 4 shows that decreasing the flow would have this effect since A appears in the denominator of the exponent; the flow must therefore be kept at its maximum value and fractionating the inlet down the length of the spaces, in an attempt to get more uniform temperature, would cause a very substantial increase in relative humidity.

IN BRIEF . . .

Power Shortages.—The War Production Board set up last month machinery to handle power shortages wherever and whenever they occur in this country. It issued an order (L-94) of nationwide application which may affect every user of electric power in the country. Purpose of the order is to assure a steady flow of power to war industries and essential civilian services by curtailing non-essential uses. Curtailment of electricity for regular consumers, however, will not take place until an area becomes a power shortage area.

Specialties.—Schedule No. VIII to limitation order L-42 requires the simplification of vacuum and vapor heating specialties after June 15. These include various types of traps, strainers, and valves. It is estimated by the plumbing and heating branch of the War Production Board that this schedule will save about 100 tons of stainless steel, and approximately 850 tons of copper and copper base alloy.

Steel and Copper.—A sweeping conservation order (M-126) issued last month by the War Production Board forbid the use of iron and steel in a list of some 400 items, including heating, piping and air conditioning equipment. Any appeals from provisions of the order must be made on form PD-437. . . . Another amendment to the copper order issued last month prohibited the use of copper and its alloys in over 100 items.

ANALYSIS OF PERIODIC LOADS IN COMPOSITE WALLS

By B. F. Raber and F. W. Hutchinson

Professor and Assistant Professor, respectively, of Mechanical Engineering, University of California

THE GRAPHICAL method of determining the temperature at any point, for any time, in a homogeneous wall subject to periodic variation of the outside air temperature [see April, 1942, HPAC, pp. 241-242] can be readily extended to composite walls consisting of any number of homogeneous sections. The example illustrated on these pages is for a particular wall, but the routine procedure of setting up the problem and of carrying out the graphical construction is explained in general terms permitting direct application to any other composite wall.

Fig. 1—The actual wall selected as an example consists of an inside surface of ½ in. plaster on ⅜ in. plaster board, an outside surface of ¾ in. pine siding on 1 in. pine sheathing, 3⅝ in. of rock wool in the wall and film coefficients of 1.6 and 6.0 Btu per hr per sq ft per F on the inside and outside surfaces respectively. Since the plaster and plaster board have approximately the same characteristics, these two materials can be considered as one ⅞ in. thickness of plaster; similarly the siding and sheathing can be regarded as one 1¾ in. section of pine. Values of the conductivity, k, specific heat, c, and density, ρ, are shown on the figure.

Fig. 2—The inside and outside reference lines I and O, and points on these lines representing air temperatures at time intervals of ½ hr, are established (see dotted lines) by transfer from the temperature-time curves for inside and outside air temperature variation over the 24 hr cycle. In this example the inside temperature is assumed constant at 70 F while the outside temperature varies sinusoidally (or by exact data from weather bureau) from a maximum of 70 F at 2 p.m. to a minimum of 30 F at 2 a.m. The

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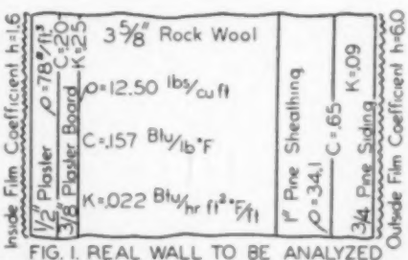


FIG. 1. REAL WALL TO BE ANALYZED

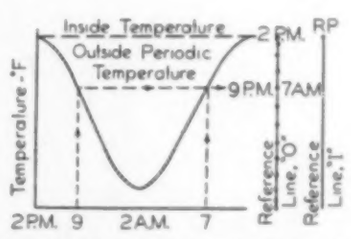
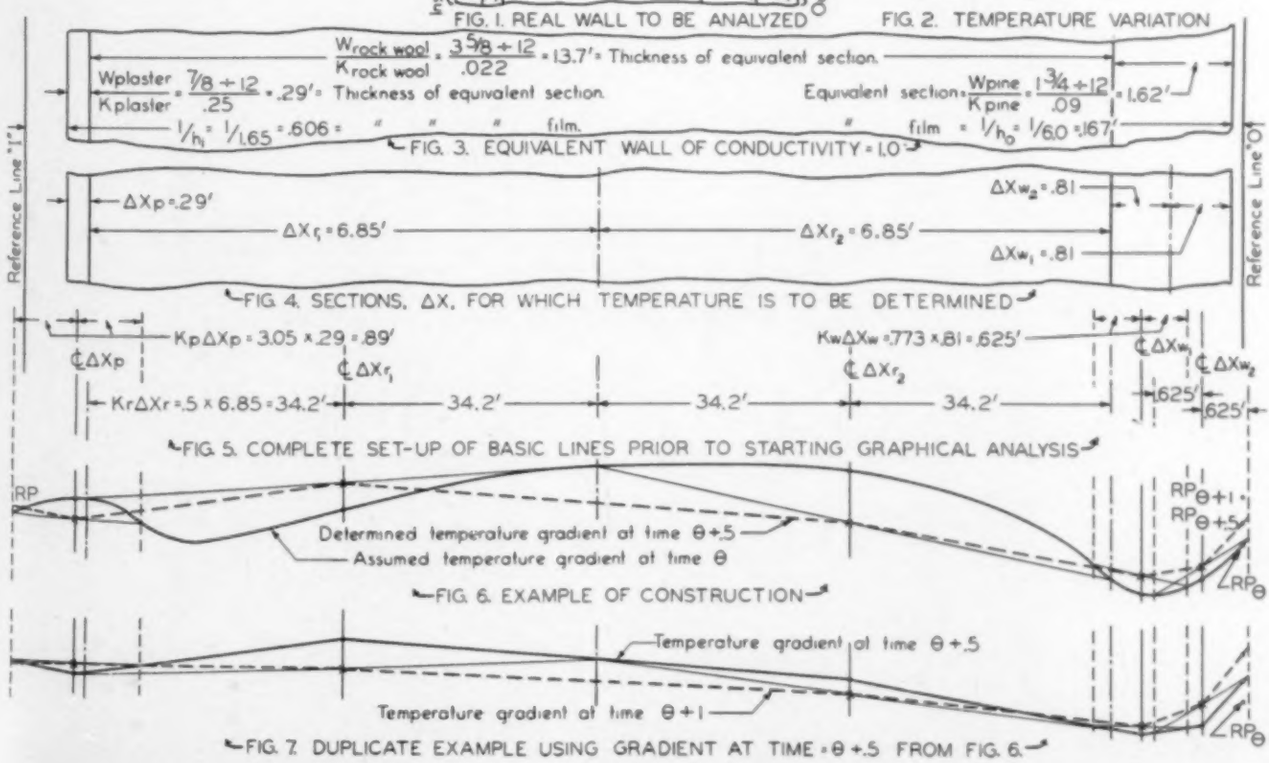


FIG. 2. TEMPERATURE VARIATION


Wrock wool / Krock wool = 3 5/8 + 12 / .022 = 13.7' = Thickness of equivalent section.
Wplaster / Kplaster = 7/8 + 12 / .25 = .29' = Thickness of equivalent section.
Equivalent section: Wpine / Kpine = 1 3/4 + 12 / .09 = 1.62'
1/hi = 1/1.65 = .606 = " " " film.
" film = 1/ho = 1/6.0 = .167'
FIG. 3. EQUIVALENT WALL OF CONDUCTIVITY = 1.0
Reference Line "I"
Reference Line "O"
ΔXp = .29'
ΔXc = 6.85'
ΔXr2 = 6.85'
ΔXw2 = .81
ΔXw1 = .81
FIG. 4. SECTIONS, ΔX, FOR WHICH TEMPERATURE IS TO BE DETERMINED
Kp ΔXp = 3.05 x .29 = .89'
Kw ΔXw = .773 x .81 = .625'
¢ ΔXp
¢ ΔXr1
¢ ΔXr2
¢ ΔXw1
¢ ΔXw2
Kr ΔXr = .5 x 6.85 = 34.2'
34.2'
34.2'
34.2'
.625'
.625'
FIG. 5. COMPLETE SET-UP OF BASIC LINES PRIOR TO STARTING GRAPHICAL ANALYSIS
RP
Determined temperature gradient at time Θ+.5
Assumed temperature gradient at time Θ
RPΘ+1
RPΘ+.5
RPΘ
FIG. 6. EXAMPLE OF CONSTRUCTION
Temperature gradient at time Θ+.5
Temperature gradient at time Θ+1
RPΘ
FIG. 7. DUPLICATE EXAMPLE USING GRADIENT AT TIME = Θ+.5 FROM FIG. 6.

slope, range and relative position of the two curves selected for this example are purely arbitrary.

Fig. 3—Starting with the inside reference line, an equivalent film and equivalent section of wall are laid off to the right such that a new wall is established of equivalent uniform conductivity equal unity. The purpose of this construction is to eliminate the change in direction of the temperature gradient which occurs at interfaces in the actual wall.

Fig. 4—Selecting a time interval, $\Delta\theta$, calculate the equivalent increment of wall width from the equation,

$$\Delta x = \sqrt{\frac{2\,(\Delta\theta)}{\rho c k K}}$$

where K is an arbitrary constant usually taken as unity for the first approximation. Then for the rock wool

$$\Delta x_r = \sqrt{\frac{2\text{x}0.5}{12.5\text{x}0.157\text{x}0.022\text{x}1}} = 4.82 \text{ ft}$$

But 4.82 divided into the equivalent width, 13.7, does not go an even number of times (2.84). Then choose a value for Δx such that $2(\Delta x) = 13.7$ and solve for K_r,

$$K_r = \frac{2\,(\Delta\theta)}{\rho c k\,(\Delta x)^2} = \frac{2\text{x}0.5}{12.5\text{x}0.157\text{x}0.022\text{x}6.85^2}$$
$$= 0.52 = \text{approximately } 0.5$$

Similarly choose one and two sections for the plaster and wood respectively with consequent values of $K_p = 3.05$ and $K_w = 0.773$. Lay off the sections Δx_p, Δx_r, Δx_w.

Fig. 5—The temperature is to be determined at the center of each Δx section. Draw in the center lines. From each center line lay off the distance $K\,(\Delta x)$ to right and left and draw vertical construction lines.

Fig. 6—Assume a temperature gradient at any time θ. This gradient intersects each of the two construction lines associated with every center line. Connect each such pair of intersections and note the point at which this line intersects the center line; that point represents the temperature on center line ½ hr later. Similarly determine temperatures at time $\theta + \frac{1}{2}$ on all center lines and connect these points with straight lines. A straight line is also drawn from point on center line farthest left to the unchanging inside reference point RP and from center line farthest right to outside reference point for $RP\theta+\frac{1}{2}$.

Fig. 7—Starting with the temperature gradient at $\theta+\frac{1}{2}$ (from Fig. 6) determine gradient at time $\theta+1$ (by same method as in above paragraph.) This procedure is repeated for 24 hr cycle. If at that time the new gradient superimposes on assumed gradient the assumption was correct and a solution has been realized; if not, repeat. Usually one, or at most two, cycles will suffice to obtain a solution.

Fig. 8—This figure gives the complete solution of the example considered. Solutions for other walls can be obtained in a similar way and can be extended to other temperature ranges or to summer loads by the methods already explained in the April HPAC.

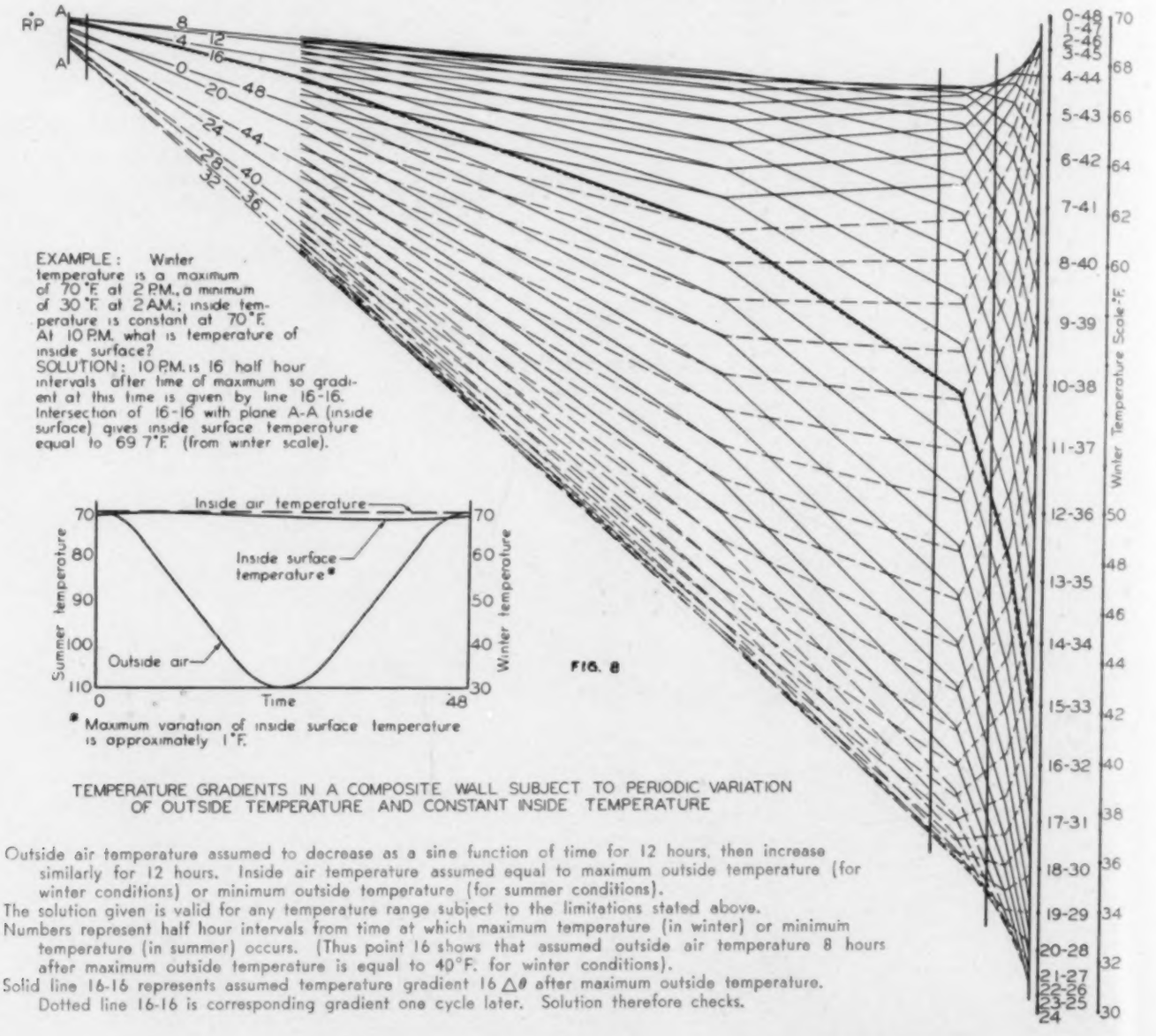


FIG. 8

TEMPERATURE GRADIENTS IN A COMPOSITE WALL SUBJECT TO PERIODIC VARIATION OF OUTSIDE TEMPERATURE AND CONSTANT INSIDE TEMPERATURE

Outside air temperature assumed to decrease as a sine function of time for 12 hours, then increase similarly for 12 hours. Inside air temperature assumed equal to maximum outside temperature (for winter conditions) or minimum outside temperature (for summer conditions).

The solution given is valid for any temperature range subject to the limitations stated above.

Numbers represent half hour intervals from time at which maximum temperature (in winter) or minimum temperature (in summer) occurs. (Thus point 16 shows that assumed outside air temperature 8 hours after maximum outside temperature is equal to 40°F. for winter conditions).

Solid line 16-16 represents assumed temperature gradient 16 $\Delta\theta$ after maximum outside temperature. Dotted line 16-16 is corresponding gradient one cycle later. Solution therefore checks.

Heating · Piping and Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

Message From Minnesota

The situation of our Country is such that every one of us is devoting all his energies and time toward winning this war. This adds an extra strain on our mental and physical facilities and requires that some diversification or form of recreation is essential if we are to keep physically fit in order to meet the requirements of our daily work. Therefore, the meeting as sponsored by our Society should be attended in full by our membership.

Let's make it a date at St. Paul, June 15-17.

Features for Future Issues

Effect of Insulation in Four Identical Dwellings, by D. B. Anderson

Effect of Vanes in Reducing Pressure Loss in Elbows in 7-Inch Square Ventilating Duct, by M. C. Stuart, C. F. Warner and W. C. Roberts

Overloading of Viscous Air Filters During Accelerated Tests, by Frank B. Rowley and Richard C. Jordan

Simultaneous Heat and Vapor Transfer Characteristics of an Insulating Material, by F. G. Hechler, E. M. McLaughlin, and E. R. Queer

Trend Curves for Estimating Performance of Panel Heating Systems, by B. F. Raber and F. W. Hutchinson

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The Specific Heat of Thermal Insulating Materials

By Gordon B. Wilkes* and Carl O. Wood**, Cambridge, Mass.

SUMMARY—In order to provide data which has thus far been lacking, the authors have experimentally determined the specific heat of fifteen of the more common insulating materials. The mean specific heats were determined for the temperature range within which the materials are generally used. This range was from 80 to 212 F for the organic insulators and from 80 to 1350 F for the higher temperature materials. True specific heats were calculated for the materials used above 212 F. A discussion of the importance of heat capacity as applied to thermal insulation is included.

Introduction

THE RESISTANCE to heat flow of the various thermal insulating materials under steady state conditions has been known for some time. In this case the conductivity value, k, and known conditions of temperature, thickness, and area are sufficient to determine the rate of heat flow through a particular material. However, in the unsteady state the specific heat and density are also important factors governing the performance of the materials, and in many cases they may be fully as important as the conductivity.

For a number of years the use of insulating firebrick of low heat capacity to replace regular firebrick in kilns and furnaces that are intermittently heated, has shown decisively that large savings in fuel and time can be obtained. McCullough[1] gives the results of tests on two furnaces, identical except for the type of lining, which were heated empty in order to compare the fuel requirements and heating time. The furnace with an insulating firebrick lining was heated to 2000 F with only one-third the quantity of gas required to heat the furnace lined with regular firebrick to the same temperature. Also the furnace lined with insulating brick reached 2000 F in approximately ⅓ the time required for the other furnace.

In ovens and furnaces operated at lower temperatures (less than 1400 F) similar savings can be obtained. As an illustration, consider an oven insulated (1) with 4 in. of glass wool or (2) with 4 in. of 85 per cent magnesia. The oven, if maintained at a constant temperature of 500 F, will then lose very nearly the same amount of heat whether insulated as in (1) or (2). This statement is based on the premise that the k values are approximately equal for the mean temperature between 80 F and 500 F. Although this assumption is not true for all types of the two insulations, it will be satisfactory for this comparison.

If the period during which the oven is raised in temperature is considered, then the heat quantities involved in the two arrangements will be substantially different. Also the time required to reach equilibrium will be considerably more in the second case than in the other.

Assume the oven insulated as in (1) to be heated from 80 F to 500 F. Taking the mean specific heat of glass wool to be 0.234 as determined in this investigation and assuming a density of 3 lb/ft³, then the heat required per cubic foot of material will be:

$$\frac{(500\text{-}80)}{2} \times 0.234 \times 3 = 148 \text{ Btu/ft}^3$$

Similarly the oven insulated with 85 per cent magnesia will require

$$\frac{(500\text{-}80)}{2} \times 0.284 \times 15 = 895 \text{ Btu/ft}^3$$

to heat the insulation.

Thus the heat requirements for the two insulations are in the ratio of 1 to 6. The heat capacity of the framework would reduce the ratio of total heat requirements according to the quantity of material in the framework and the type of construction. In no case, however, would this factor reverse the ratio since the same supporting structure was assumed in each case.

In comparing the times required to heat the apparatus to the desired temperature, the thermal conductivity must also be considered. The conductivity divided by the product of specific heat and density ($k/c\rho$) gives a term commonly referred to as *thermal diffusivity*. For materials of the same thickness, the heating time is inversely proportional to the diffusivity. For materials of the same diffusivity, the heating time is directly proportional to the square of the thickness. Consequently, for any material of any thickness, the controlling factor in heating time is $\frac{1}{k/c\rho} \times L^2$ or $\frac{c\rho L^2}{k}$ where L is the thickness.[2]

Since the two installations are the same thickness, however, the thermal diffusivity alone can be used for comparison of heating times.

Thus glass wool with a mean conductivity between 80 and 500 F of 0.44 Btu, hr^{-1}, ft^{-2}, F^{-1}, in. has a diffusivity of $0.44/0.234 \times 3 \times 12$ or 0.052 ft²/hr. In order to have consistent units, the k factor of 0.44 is divided by 12 so that the new k is expressed in Btu, hr^{-1}, ft^{-2}, deg F^{-1}, ft.

The conductivity of 85 per cent magnesia can also be taken as 0.44. Then the diffusivity will be $0.44/0.284 \times 15 \times 12$, or 0.0086 ft²/hr.

It is seen that the oven insulated with glass wool will reach 500 F in 0.0086/0.052 or approximately 1/6 of the time required for the magnesia installation.

These examples show that by consideration of heat capacity in addition to other properties a more satisfactory selection of insulation can often be made.

In order to provide data on the specific heat of thermal insulation an investigation has been carried

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[1] Insulating Firebrick for Kiln Construction, by J. D. McCullough. (*Bulletin American Ceramic Society*, 20 [12] 1941, pp. 437-442.)

For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, St. Paul, Minn., June, 1942.

[2] *Heat Transmission*, by McAdams, Chapter II, 1933.

out on some of the common materials. Little has been done along this line in the past. Until recently there has been very little interest among users of insulation either because they did not know the effect of heat capacity or they relied on past experiences. Furthermore, the very nature of the materials presents considerable difficulty in measuring the specific heat.

The usual calorimetric apparatus employs a liquid, usually water, in which a heated specimen is dropped. The temperature rise of the liquid times its specific heat and weight will then give the total heat received from the specimen. The contact between liquid and specimen permits thermal equilibrium to be established in a comparatively few minutes. The heat gained from or lost to the surroundings during this short interval will then have only a small effect on the final accuracy.

However, because of the various binders and fireproofing chemicals present in insulating materials, a chemical reaction may take place when the substance is placed in water or other liquid. This of course would change the measurement considerably. Also the fact that some of the materials will not sink, even though compressed to five or ten times their original densities, proved to be a considerable problem.

For these reasons it was found advantageous in this investigation to use a dry calorimeter[3] in which a block of metal was substituted for the liquid. The materials were compressed to pellets with five or ten times the original densities but the thermal conductivity was still relatively low. This, along with the high thermal resistance between the sample and calorimeter, hindered heat flow to a considerable extent.

In order to hold the time required to reach equilibrium to a minimum, it was necessary to use as small a sample as possible. At the same time the sample had to be large enough to maintain satisfactory accuracy in weighing. Furthermore, as it was found necessary to cover the sample with metal foil to prevent loss of material, the sample had to be large enough so that the foil heat capacity would be relatively unimportant.

Because of these limitations, the time required for the sample to reach equilibrium with the calorimeter was fixed at approximately 50 min. Although this comparatively long time is not usually satisfactory for calorimetric measurements, its effect on the final accuracy was reduced considerably by calibrating the calorimeter under practically the same conditions as in an actual determination.

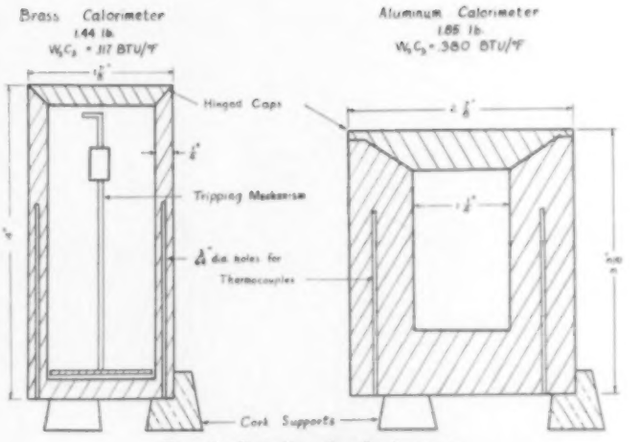

Fig. 1—Details of calorimeters

Fig. 2—The apparatus for determining the specific heat of thermal insulation

Apparatus and Method

The mean specific heat between 80 F and temperatures ranging from 212 F to 1350 F were determined experimentally for 15 materials. In addition, the variation of true specific heat with temperature was calculated for those materials which, by virtue of their use in industry, seemed to warrant determinations at more than one temperature.

The apparatus consisted essentially of a metal calorimeter complete with an *adiabatic* jacket, a potentiometer and thermocouples for measuring the temperatures of the specimen and calorimeter, and furnaces for heating the specimen to the desired temperature.

The calorimeter itself was a cylindrical block of metal with a hole drilled part way through parallel to the axis. The open end was covered by means of a hinged cap. Three small corks glued on the other end served as supports (see Figs. 1 and 2).

The temperature of the calorimeter was measured with a double copper-constantan thermocouple and a portable precision type potentiometer. The two measuring junctions of the thermocouple were placed inside small glass tubes and inserted in holes drilled into the calorimeter wall from the bottom. The reference junctions were kept in a thermos bottle filled with cracked ice and water.

The jacket consisted of two spun copper cans of unequal size, placed one inside the other so as to have a ¾-in. space between at all points except the top. The space was filled with water. The two cans were then connected across a 110 volt power line with a switch in the circuit so that the jacket could be heated as desired. In order to increase the efficiency of the jacket, the outer can was surrounded by 2 in. of insulation in a still larger container.

The temperature difference between jacket and calorimeter was indicated by means of a differential thermocouple and a spotlight galvanometer. One junction of the couple was placed inside a thin glass protection tube and immersed in the jacket water. The other was soldered to the outer surface of the calorimeter.

In determining the specific heat of a given material, a sample was first prepared by compressing it under a pressure of about 100,000 lb per square inch so as to form a solid pellet. This pellet was then cut down to the desired weight (from 2 to 15 g (grams) depending on the material and the temperature to which it was to be heated, correlated with a calorimeter temperature rise of from 4 to 7 F.) The sample was

[3]Improved Metal Calorimeter for Determination of Specific Heats of Metals, Oxides and Slags, by Grosse and Dinkler. (*Stahl und Eisen*, 47 [11] 1927, pp. 448-453.)
The Specific Heat of Magnesium and Aluminum Oxides at High Temperatures, by G. B. Wilkes. (*Journal American Ceramics Society*, 15 [1] 1932, pp. 72-77.)

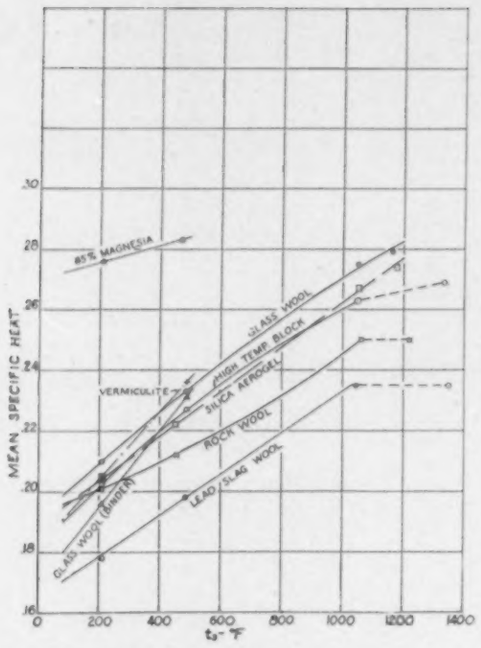

Fig. 3—Mean specific heat between t_1 and 80 F vs. t_2

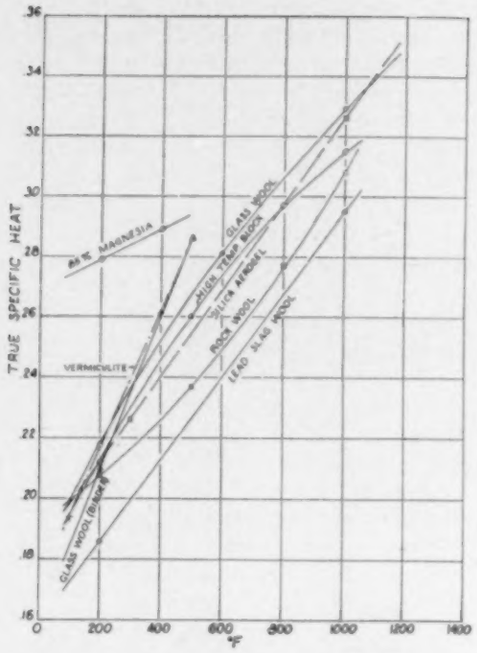

Fig. 4—True specific heat vs. temperature

Table 1—Results of All Materials Tested

Material	Mean Specific Heat[a]	Temperatures Deg Fahr.[b]		
		t_2	t_1	Mean
Lead slag wool	0.178	88	212	150
	0.198	88	484	286
	0.235	86	1044	565
	0.235	86	1350	718
Glass wool (with binder)	0.196	87	212	150
	0.231	86	490	288
Rock wool	0.201	86	212	149
	0.212	87	455	271
	0.250	87	1063	575
	0.250	85	1219	652
High temperature block	0.203	86	212	149
	0.227	90	490	290
	0.263	89	1049	569
	0.269	84	1336	710
Silica Aerogel	0.205	82	212	147
	0.222	87	455	271
	0.267	86	1056	571
	0.274	83	1177	630
Vermiculite	0.205	85	212	149
	0.236	90	490	290
Glass wool	0.210	88	212	150
	0.233	83	487	285
	0.275	87	1051	569
	0.279	87	1161	624
Corrugated asbestos paper	0.245	82	212	147
Rock wool with asphaltic binder	0.247	85	212	149
85 per cent Magnesia	0.276	88	212	150
	0.283	88	470	279
Wood fibre blanket	0.330	85	212	150
Hair felt	0.334	84	212	148
Wood fibre board	0.341	83	212	148
Expanding paper blanket	0.349	84	212	148
Cork board	0.417	87	212	150

[a] Average of several runs where more than one was made. Corresponding temperatures are also averages.
[b] Temperatures given to nearest degree.

Table 2—Constants for the True Specific Heat Equation

$$C_T = A + B\,t + C\,t^2$$

Material	A	B	C	Temp. Range Deg Fahr.
Lead Slag Wool	0.159	0.000136	0	80 to 1050
Rock Wool	0.189	0.0000738	0.0000000450	80 to 1050
Silica Aerogel	0.184	0.000142	0	80 to 1180
High temperature block	0.174	0.000202	—0.0000000611	80 to 1050
Glass wool	0.182	0.000192	—0.0000000450	80 to 1170
Glass wool with binder	0.159	0.000253	0	80 to 500
Vermiculite	0.171	0.000226	0	80 to 500
85 per cent Magnesia	0.268	0.0000533	0	80 to 470

Table 3—Specific Heat, Heat Capacity, and Thermal Diffusivity of Various Insulating Materials

Note: Density values given are only approximate. This table is intended for a rough comparison and not for a rigorous statement of properties.

Material	Mean Values Between 80 F and 212 F				
	Specific Heat	Density[a] Variation	Average Density[a]	Heat Capacity[b]	Thermal Diffusivity[c]
Expanding paper blanket	0.349	.....	1.6	0.55	0.046
Glass wool	3.210	1.5—4.5	3	0.63	0.036
Silica aerogel	0.205	.....	4*	0.82	0.020
Wood fibre blanket	0.330	2—4	3	0.99	0.023
Rock Wool	0.201	2—10*	6*	1.2	0.019
Lead slag wool	0.178	4—10*	7*	1.3	0.017
Vermiculite	0.205	8—10*	9*	1.8	0.025
Glass wool (binder)	0.196	.....	12*	2.3	0.011
Cork board	0.417	.....	8	3.3	0.0068
Hair felt	0.334	.....	11**	3.7	0.0061
85 per cent Magnesia	0.276	12—18	15	4.1	0.0086
Corrugated asbestos paper	0.245	12—22	17	4.2	0.0089
Rock wool with asphaltic binder	0.247	.....	17	4.2	0.0066
High temperature block	0.203	18—28	23	4.7	0.011
Wood fibre board	0.341	12—17*	14.5*	5.0	0.0055

[a] lb/ft³ Values are as given by manufacturers, unless specified as follows:
*From tests conducted in Heat Measurements Laboratory, M.I.T.
**Heating, Ventilating, Air Conditioning Guide, 1942, p. 95.
[b] Specific heat times average density Btu/ft³/F.
[c] Thermal conductivity divided by heat capacity, Ft²/hr.

then covered with metal foil to prevent loss of material and reweighed to determine the amount of foil. Aluminum foil was used for the low temperature runs and platinum foil for temperatures above 1100 F.

The sample was heated and maintained at a known temperature for several hours. It was then dropped into the calorimeter and the temperature rise of the latter measured.

The jacket was kept at very nearly the same temperature as the calorimeter at all times in order to reduce heat losses from the calorimeter. The average temperature difference during a run on any of the materials was only 0.05 F. Heat exchange between the two parts was further reduced by covering both the calorimeter and the inside surface of the jacket with aluminum foil.

Due to the limitations on the size of the samples, it was found necessary to use two calorimeters; one for the low temperature runs (212 F-500 F) and the other for the higher temperatures. The heat capacities of the two calorimeters were 0.117 Btu/F and 0.380 Btu/F, the smaller being made of brass, the other of aluminum.

The calorimeters were calibrated with a very small electric heater. After placing the heater inside the calorimeter and making sure that the temperature of the apparatus stayed constant, current was supplied for a period of twenty minutes, after which the calorimeter was allowed to reach equilibrium. Current and potential readings taken at one minute intervals gave the power input to the heater. The heat capacity of the calorimeter could then be calculated from the relation:

$$E\ I\ \tau \times 3.413 = W_3\ C_3\ (t_3 - t_1)$$

where: E = voltage drop across heater, volts
I = amperes through heater,
τ = time heater was on, hours,
W_3C_3 = heat capacity of calorimeter (W_3 = weight)
C_3 = specific heat, Btu/F
t_1 = initial temperature of calorimeter, F
t_2 = final temperature of calorimeter, F

Corrections for the heat capacity of the heater itself were negligible, amounting to only 0.0002 Btu/F as calculated with specific heat and weight data.

The samples were heated in three different furnaces depending on the temperature desired. A small steam jacket with radiation shield was used for heating the sample to 212 F, the sample being held in a pyrex glass test tube which hung in the steam bath. In transferring the sample from the steam bath to the calorimeter, the test tube was lifted over to the calorimeter and inverted, care being taken that no water droplets fell into the calorimeter. Due to the relatively large heat capacity of the test tube, heat losses from the sample were negligible during the transfer period. The sample was exposed to the room for only about two seconds as it dropped into the calorimeter. The steam temperature was calculated from the barometric pressure.

An electrically heated furnace was constructed for the 500 F determinations. This was small enough so that it could be lifted directly over the calorimeter for dumping the sample.

For temperatures above 500 F a larger electrically heated furnace was needed. Here again a container of relatively large heat capacity held the sample while in the furnace and during the transfer to the calorimeter. A porcelain tube sealed at one end was used in this case.

In all determinations above 212 F, the temperature of the sample in the furnace was measured with a thermocouple which laid alongside the sample.

The mean specific heat was calculated from the experimental data as follows:

$$\frac{W_1}{454}(t_3 - t_2)C_1 + \frac{W_2}{454}(t_3 - t_2)C_2 = W_3\ C_3(t_2 - t_1)$$

where: W_1 = weight of sample, grams
W_2 = weight of foil, grams
t_1 = initial temperature of calorimeter, Fahrenheit
t_2 = final temperature of calorimeter, Fahrenheit
t_3 = temperature of sample in furnace, Fahrenheit
C_1 = mean specific heat of sample between t_3 and t_2
C_2 = mean specific heat of foil between t_3 and t_2
W_3C_3 = heat capacity of calorimeter, Btu/Fahrenheit

Then solving for C_1:

$$C_1 = \frac{W_3\ C_3(t_2 - t_1) - \frac{W_2}{454}(t_3 - t_2)\ C_2}{\frac{W_1}{454}(t_3 - t_2)}$$

The results for all the materials tested are given in Table 1. Fig. 3 shows these data plotted as mean specific heat *vs.* t_3.

In order to find the variation of true specific heat, C_T, with temperature, the following relation was assumed:

$$C_T = A + B\ t + C\ t^2$$

where A, B, and C are constants. Considering the heat involved in raising one pound of material from t_2 to t_3:

$$C_1\ (t_3 - t_2) = \int_{t_2}^{t_3} (A + B\ t + C\ t^2)\ dt$$

Or carrying out the integration gives:

$$C_1\ (t_3 - t_2) = \left[A\ t + \frac{B\ t^2}{2} + \frac{C\ t^3}{3}\right]_{t_2}^{t_3}$$

If determinations of C_1 are made for three different temperature differences then appropriate values of t_2, t_3 and C_1 can be substituted in the equation to obtain three simultaneous relations and the constants A, B, and C determined.

In several cases the relation between C_1 and $(t_3 - t_2)$ proved to be a straight line so that only two equations were needed.

The constants found in this way are listed in Table 2. Fig. 4 shows true specific heat plotted against temperature.

Discussion of Results

Referring to Fig. 3 it is seen that above 1050 F the mean specific heat of the high temperature block, rock wool and lead slag wool change markedly. This is due to a chemical change which is further evidenced by a decided change in color. This variation in specific heat is not important, however, since only a very small layer of the insulation would actually reach a temperature above 1050 F.

Since the performance of materials in unsteady heat flow conditions involves also the density and conductivity, a fair comparison is not obtained from specific heat data alone. As the density of most thermal insulating materials is a highly variable quantity, it would be impossible to rigidly compare their heat capacities and thermal diffusivities.

However, as a matter of interest the mean heat capacity values between 80 and 212 F are listed in

Table 3 in the order of increasing value. Also the corresponding thermal diffusivities are given. The densities and conductivities used were obtained from manufacturers' circulars where possible. Otherwise values were taken from tests conducted at the Massachusetts Institute of Technology or from the HEATING VENTILATING AIR CONDITIONING GUIDE, 1942.

Assuming the same thickness of material in all cases, Table 3 shows that the expanding paper blanket insulation will not only require the least amount of heat to raise its temperature from 80 to 212 F but also will heat up faster than any of the materials tested. Of the inorganic insulators tested, glass wool has the lowest heat capacity and the highest diffusivity.

Of course in many instances a high heat capacity and low diffusivity is desired. An annealing furnace, for instance, might require slow cooling if material is left in it after being heated to the desired temperature. In this case 85 per cent magnesia or high temperature block insulation might be the best choice.

Allcut[4] notes several instances in Canada where the use of thick insulation to obtain a large heat capacity has been very satisfactory in maintaining constant temperatures and low fuel costs.

He cites the example of a hospital in Prince Edward Island which utilized insulation 10 in. thick and required less than half the heating capacity of similar hospitals with ordinary construction. This saving was attributed to the effect of time lag in alternately storing and returning heat from both the heating system and solar radiation.

An instance where several unheated packing sheds and storage buildings utilized the effect of heat capacity was also mentioned by Professor Allcut. Because of the large heat capacity of the walls, the buildings have been consistently maintained at temperatures above 30 F even though outdoor temperatures often fall below zero.

[4]Properties of Heat Insulating Materials, by E. A. Allcut. (*The Engineering Journal*, 24 11 1941, pp. 514-524.)

Table 4—Mean Specific Heat of Materials Tested

Materials	Mean Specific Heat	Mean Temp. Deg Fahr.	Results of This Investigation	
			Materials	Mean Specific Heat
Slag wool[a]	0.17	77	Lead slag wool	0.17
Baked cork[a] (slab)	0.43	77	Cork board	0.417[f]
Granulated cork[a]	0.44[e]	77	Cork board	0.417[f]
Cork (expanded but not impregnated)[b]	0.433	91	Cork board	0.417[f]
Natural cork[b]	0.419	124	Cork board	0.417[f]
Kapok fibre[b]	0.320	65		
Cellulose[c]	0.319—0.327	*	Organic insulators	0.330—0.417[f]
Asbestos[c]	0.25	*	Corrugated asbestos paper	0.245[f]
Glass[d]	0.14—0.23	*	Glass wool	0.210[f]

[a]Heat Insulators in *Food Investigation*, Special Report No. 35, National Physical Laboratory, London (1929).

[b]Specific Heat of Building and Insulating Materials, by M. Kinoshita. (*Journal Society of Chemical Industries*, London, 36, 1917, 597).

[c]International Critical Tables, Vol. II, 237 (1927).

[d]Ibid, Vol. II, 93.

[e]Material contained 3.4 per cent moisture.

[f]Mean temperature 150 F.

[*]Temperatures not given.

Comparison of the specific heat values with those of other observers shows good agreement. This correlation, however, was possible for only a few of the materials because of the meager data in the literature.

Since heat insulators are seldom pure substances, the small variations in values of different observers are likely due mainly to slight differences in composition.

In Table 4 are listed values obtained from different sources along with those obtained on similar materials during the course of this investigation.

Precision

The precision obtained was ± 1 per cent as derived by the following considerations:

The temperature rise of the calorimeter was measured with a double copper-constantan thermocouple. Thus, since the potentiometer could be read to ±0.001 millivolts, the actual value was known to ±0.0005 millivolts. Therefore, the third decimal place was fixed with sufficient accuracy, the temperature rise being greater than 0.100 mv. (single couple).

The temperature of a sample in the furnace was measured with a single copper-constantan thermocouple at the lower temperatures, and a single chromel-alumel couple above 1000 F. Calibrations for both couples were known to well within ± 1 per cent error. Since the couple lay alongside the sample and touching it inside the heater, and since sufficient time was given for the whole to come to equilibrium, there is little chance for error in the assumption of a ± 1 per cent accuracy for this measurement.

The temperature of the steam bath was known to well within ± 1 per cent.

The time required to transfer a sample from furnace to calorimeter was only two or three seconds. In this short interval, the quantity of heat loss was negligible. In all runs the sample was held in a container of relatively large heat capacity, while in the furnace and until dropped into the calorimeter. The distance through which the sample fell was only about six inches.

The heat losses from the calorimeter to the surroundings were reduced to a minimum by means of the *adiabatic* jacket and aluminum foil as explained previously. The effect of these losses was further reduced by the calibration procedure.

Check runs were made on all but six determinations in which cases the values fell on the curve of mean specific heat vs. t_a

The weight of the samples was measured to ± 1 per cent.

The heat capacity of the foil amounted to no more than 4 per cent of that of the sample plus foil. The value of that quantity could therefore be in error by as much as 25 per cent without affecting the results.

Acknowledgment

The authors are indebted to L. R. Viäney for his help and suggestions on the construction of the apparatus.

ST. PAUL—

JUNE 15-17

SEMI-ANNUAL MEETING 1942

Performance of a Forced Warm-Air Heating System as Affected by Changes in Volume and Temperature of Air Recirculated

By A. P. Kratz* and S. Konzo**, Urbana, Ill.

SUMMARY—**This two-year investigation revealed seven significant factors in the performance of a forced warm-air heating system under stoker operation. The data developed will be helpful to the designing engineer in selecting and checking design conditions for a plant, estimating the effect on comfort of varying rates of air change, intermittent operation, air deliveries, duct system and room temperature gradients.**

Introduction

CONSIDERABLE FLEXIBILITY exists in the choice of air volumes and air temperatures to be used in the design of a forced warm-air heating system. For a given structure, the design of the duct system may be based on the use of either a large volume of air at a relatively low register air temperature or a smaller volume of air at a higher register air temperature. Furthermore, after the system is installed, satisfactory performance may be obtained even though the actual combinations of air volumes and register air temperatures encountered deviate widely from those assumed for the design of the duct system.

This flexibility in conditions affecting both the design and the operation of a forced warm-air heating system may be considered as an advantageous feature, inasmuch as any reasonable deviations in the operating conditions from those assumed for the design can be readily compensated for by relatively minor changes in the plant. On the other hand, this extreme flexibility in the choice of both design and operating conditions has resulted in the development of many *rule of thumb* methods and unconfirmed theories.

Proper development of any method of heating which is undergoing continual evolution and improvements in technique, requires that basic facts be substituted for *rules of thumb*. One of the prevailing theories of design is that in the case of a forced warm-air system installed in a well insulated structure, a certain minimum number of air recirculations, commonly referred to as air changes, is essential for satisfactory operation. For the purpose of determining whether or not the plant performance as affected by independently varying the quantity and temperature of the air circulated had an adverse effect on the environment produced, several series of tests were conducted in the Warm-Air Research Residence during the winter seasons of 1939-1940 and 1940-1941. In the first season the tests were made with the 12 warm-air registers located in the baseboard, and in the second season the tests were repeated with the same number of warm-air registers located in the sidewalls and approximately 7 ft above the floor.

In order to make direct comparisons between the results obtained with the various methods of operation, only one change at a time was made in the items specifically under observation; namely, register location, air volume, or the setting of the bonnet thermostat.

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For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, St. Paul, Minn., June, 1942.

Description of Research Residence and Heating Equipment

The Research Residence and the forced-air heating plant have been completely described in a previous publication.[1] The Residence is a three-story structure of standard frame construction with side walls and ceiling fully insulated[2] with mineral wool insulation. The total space heated, including a sun room but not including the basement, amounted to 17,540 cu ft. All of the 50 windows on the three stories of the Residence, with the exception of two small quarter-round windows in the east dormitory, were provided with tightly fitting storm sash. The calculated heat losses were approximately 51,100 Btu per hour at an indoor-outdoor temperature difference of 80 F. The Residence is completely furnished, and during the heating season it was occupied by five people.

The heating plant consisted of a cast-iron, circular-radiator warm-air furnace used in connection with a forced-air heating system. The furnace was fired by means of a coal stoker of the underfeed type. The fuel burned was 1-in. x 10-mesh stoker coal, washed and oil treated, and was obtained from Saline County, Ill.

The heating plant was controlled by means of a room thermostat operating to start and stop the stoker motor, and to start and stop the circulating fan. This room thermostat was of the heat-anticipating type, and was used in conjunction with two bonnet thermostats which served as high and low limit controls for the temperature of the air in the furnace bonnet. When

[1] University of Illinois, *Engineering Experiment Station Bulletin No. 266*, by A. P. Kratz and S. Konzo, 1934.

[2] Effect of Insulation of Plant Performance in the Research Residence, by A. P. Kratz and S. Konzo. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May, 1941, p. 318-324.)

Table 1—Sizes of Warm Air Registers

Room	Baseboard[a] Registers, Inches	High Sidewall Registers, Inches	Stack Size Above Baseboard Register, Inches
Living	12x6	14x5	3¼x14
Hall	14x4	14x4[b]	3¼x14
Dining	12x6	12x4	3¼x12
S. Sunroom	12x10	12x4	3¼x12
N. Sunroom	12x10	12x4	3¼x12
Kitchen	14x10	14x4	3¼x14
E. Bedroom	12x6	12x6	3¼x12
S. W. Bedroom	12x6	12x4	3¼x12
Bath	10x8	10x4	3¼x10
N. W. Bedroom	12x10	12x4	3¼x12
E. Dormitory	12x10	10x4[b]	3 x10
W. Dormitory	12x10	10x4[b]	3 x10

[a] Description given in University of Illinois, Engineering Experiment Station Bulletin No. 318, pages 15 and 16.
[b] Registers approximately 6 ft above floor, all others 7 ft above floor.

Table 2—Summary of Test Series

Series No.	Season	Register Location	Air Volume, cfm	Fan Cut-out Point, F	
2-39	1939-40	Baseboard	1675	125	
4-39	1939-40	Baseboard	1300	125	
5-39	1939-40	Baseboard	800	125	150[b]
1-40	1940-41	High wall	1675	125	
7-40	1940-41	High wall	1300	125	
5-40	1940-41	High wall	800	125	154[b]
4-40	1940-41	High wall	800		137[b]
6-40	1940-41	High wall	800		179[b]
3-40[a]	1940-41	High wall	800		137[b]

[a] Full register opening resulting in average register air velocity of approximately 290 fpm, whereas all other tests with high wall registers were made with 600 fpm register air velocity.
[b] Mean bonnet air temperature determined by average of maximum and minimum values. For the tests with 800 cfm the mean bonnet temperature is of greater significance than the fan cut-out point as determined from the setting of the bonnet thermostat.

the room thermostat operated to start the fan and stoker, one bonnet thermostat (fan control) prevented the fan from starting unless the bonnet temperature was above 125 F, except for the series of tests in which the bonnet thermostat settings were changed in order to determine the effect of varying the air temperature. Similarly for all tests, except those in which the bonnet thermostat settings were changed, the other bonnet thermostat (limit control) prevented the stoker from starting unless the bonnet temperature was below 150 F. Both the fan control and limit control were incorporated in one container and the adjustment of the thermostat indicator changed the setting of both controls simultaneously. The temperature differential between the fan cut-in point and the fan cut-out point was approximately 40 F.

During the season of 1939-1940 the 12 warm-air registers in the Residence were located in the baseboard and were of the sizes shown in Table 1. The registers were equipped with directional vanes and directed the air sidewise and parallel to the floor. Due to the fact that the register locations were dictated by the location of warm-air stacks which had originally been installed in the Residence in conformity with certain structural features, the arrangement cannot necessarily be considered as the best for each room. An improvement in the air distribution might have resulted had it been possible to use the most favorable location of the register in each case, but tests to verify such possibilities were not considered as within the scope of the present investigation.

During the season of 1940-1941 a stack extension was added to an opening in the top of each of the baseboard stackheads and a new stackhead was located at the top of the extension and approximately 7 ft above the floor. Sidewall registers, of the sizes shown in Table 1, were installed with the register faces flush with the plaster walls. The registers directed the air parallel to the floor with no sidewise deflection. The nearest commercial sizes of sidewall registers were selected, consistent with a register air velocity of approximately 600 fpm based on a total air recirculation of 1675 cfm. The sidewall registers were considerably smaller than the corresponding baseboard registers for which the register air velocity averaged approximately 300 fpm. During the entire series of tests no changes were made in the return air system, which included grilles in the living room, hall, and sun room.

Method of Conducting Tests

The average of the air temperatures in all of the rooms of the house was maintained at 72 F at the 60-in. level both day and night. Observations of weather, indoor room air temperatures, room relative humidities, and other incidental data were made daily at 7:00 a. m., 11:00 a. m., 4:00 p. m., and 10:00 p. m. Complete data were obtained for each 24-hour test period on the fuel consumption, weight of ash and clinkers removed, the total integrated time of operation of the fan and of the stoker, the total electrical input to the fan and stoker motors, and the total number of *on-periods* of both the circulating fan and the stoker. Daily observations were made of the volume of air circulated, and the filters were cleaned with sufficient frequency to maintain the air volume constant. In addition, continuous records of temperatures, CO_2, and the index of smoke density were obtained for each 24-hour period. Each day at 11 a. m., the clinkers were removed, the fuel bed was levelled, and the hopper was filled with coal. During extremely mild weather no attention was given to the fuel bed or the coal in the hopper, except as required every two or three days. By means of the balanced check damper, the draft in the smoke pipe was maintained at approximately 0.05 in. of water. For each series of tests, data were obtained over a wide range of outdoor weather conditions.

For the series of tests in which baseboard registers were used, the full-sized register openings were maintained with the three quantities of air circulated, and the register air velocities decreased as the quantity of air circulated was decreased, varying from a maximum of 300 fpm to a minimum of 140 fpm. On the other hand, the series of tests in which high sidewall registers were used, with the exception of series 3-40, a register velocity of 600 fpm was maintained regardless of the total quantity of air circulated. This was done by first adjusting the dampers in the duct system to obtain the volume of air required to heat each room, and then adjusting the size of the register opening by partially covering it with a plate so that a register air velocity of 600 fpm was obtained. The series of tests run are shown in Table 2.

The air delivery of the fan, listed in Table 2, was measured by means of a Pitot tube, placed in the return duct and calibrated in position. For the purpose of calibration, prior to the start of the heating season the entrances to the outlet ducts were blocked and all of the air delivered by the fan was passed through the furnace and out of a 15-in. diameter calibrating duct connected to the outlet side of the furnace casing. Twelve-point velocity traverses were made at a measuring station located in the calibrating duct, and the air volumes, as determined by the traverse, were correlated with the readings of the fixed Pitot tube located in the return duct. When the calibration was completed, the calibrating duct was removed and the entrances to the trunk ducts were opened in order to make the furnace ready for service. By adjusting the fan speed and the position of the dampers on the intake side of the fan, air volumes of 1675 cfm, 1300 cfm, and 800 cfm were readily obtained.

Results of Tests

Variations in Air Volume Delivered from Baseboard Registers (Series 2, 4, 5-39): For each series of tests, data were obtained over a wide range of outdoor weather conditions, and were plotted as points on graphs in which the abscissae were indoor-outdoor temperature differences. A curve was drawn through the points representing each series of tests, and the values obtained from the curves at indoor-outdoor temperature differences of 34 F and 55 F were selected as typical of average and relatively cold winter days respectively. The values so obtained have been plotted in Fig. 1.

In Fig. 1 the circled points and the full line curves show the significant results obtained for baseboard registers with various air deliveries on a day in which the indoor-outdoor temperature difference was 55 F. The top set of curves, showing the maximum and minimum values of the bonnet air temperatures, indicate that, although no change was made in the setting of the bonnet thermostat, the range of bonnet air temperatures was slightly higher with low air deliveries. This may be accounted for by the fact that the rate of heat evolution in the furnace was always about the same, even though the rate of heat transfer from the furnace to the circulating air was slightly less with low air deliveries than it was when higher air deliveries were used. The increase in the range of the bonnet air temperatures accompanying low air deliveries was of the order of only 10 F, and hence was not sufficient to result in any increases in basement temperature or fuel consumption, or in any temperature unbalance in the rooms.

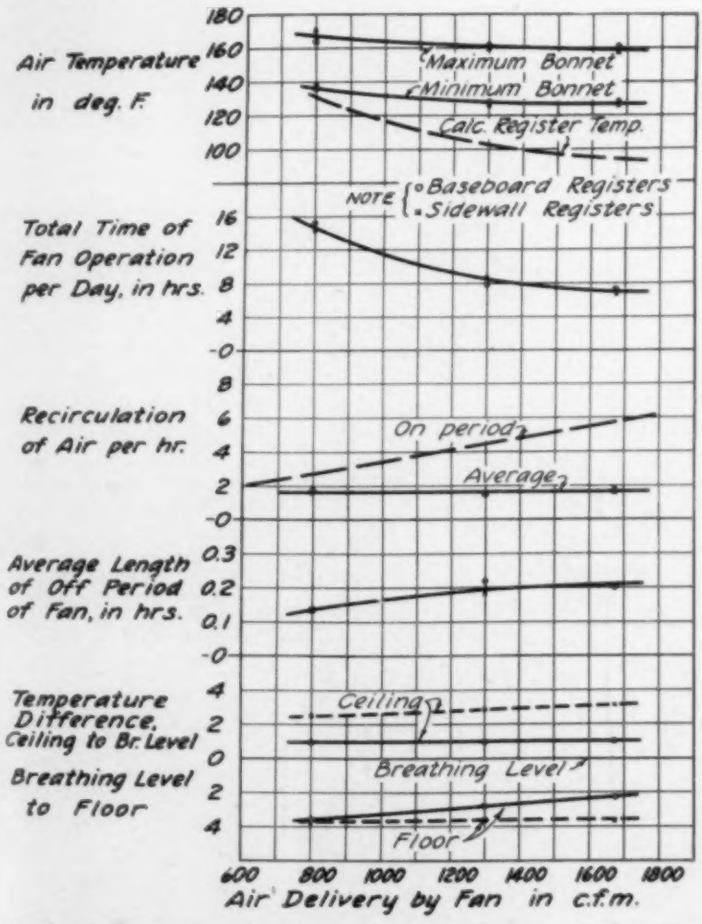


Fig. 1—Performance of baseboard and sidewall registers with various air deliveries
Temperature difference of 55 F indoor to outdoor bonnet setting 125 F. Stoker-fired furnace

The curve designated as calculated register temperature represents the register air temperatures that would have been used in the design of different forced-air systems for the Research Residence, based on the calculated heat losses for the Residence and the various air deliveries shown in Fig. 1. In order to offset the heat loss from the rooms, either a large air delivery accompanied by low register air temperature or a small air delivery accompanied by a high register air temperature would be used. In this case, in which the Residence was insulated, air deliveries of 1675 cfm, 1300 cfm, and 800 cfm were accompanied by design register air temperatures of 94 F, 103 F, and 131 F, respectively, corresponding to calculated mean bonnet air temperatures of 102 F, 115 F, and 151 F. For the tests shown in Fig. 1 the setting of the bonnet thermostat was not changed, and hence the actual mean bonnet air temperatures were not adjusted to conform to the lower calculated bonnet air temperatures when large air deliveries were used. With the greater air deliveries, the calculated mean bonnet air temperatures were lower than the actual, and hence the total time of fan operation was not as great as it would have been if design conditions had been met. In this respect the results tended to favor to some extent the operation with low air deliveries. However, satisfactory operation was obtained with an air delivery of 1675 cfm, indicating the flexibility of the plant even under conditions when the operating air temperature deviated from the design air temperature.

The total time of fan operation per day, as indicated in the second set of curves from the top of Fig. 1, was approximately twice as great for 800 cfm delivery as it was for 1675 cfm delivery. However, the electrical input to the fan, expressed in terms of watt-hours per day, was slightly less for the lower air de-

livery. Hence reducing the speed of the fan resulted in an increase in operating time with no increase in electrical input.

The circulation of 1675 cfm during the *on-period* operation of the fan was equivalent to 5.73 recirculations per hour. If no circulation was assumed to occur during the *off-periods* of the fan, a weighted average value for both *on-periods* and *off-periods* of 1.75 recirculations per hour was obtained. Corresponding values for 800 cfm air delivery were 2.74 recirculations per hour for *on-period* fan operation alone and 1.75 recirculations per hour for the weighted average of *on-periods* and *off-periods*. Hence, the average air recirculation per hour remained constant over a very wide range of air deliveries by the fan, while the *on-period* delivery increased from 2.74 to 5.73 recirculations per hour.

Comfort conditions in the room are improved by any decrease either in the length of *off-periods* of the fan or in the temperature difference between the floor and the breathing level. The lower two sets of curves in Fig. 1 show that an air delivery of 800 cfm resulted in shorter *off-periods* for the fan, but in a slightly larger temperature differential than those occurring with an air delivery of 1675 cfm. The temperature differences shown are the averages for 10 rooms in the Residence, including the relatively large differences obtained in the sun room and the two dormitories. If the open areas of the baseboard registers had been reduced as the air deliveries were lowered, so that a constant register velocity was maintained irrespective of the fan delivery, it is possible that the floor temperatures would not have decreased to the extent shown. In any case, the differences obtained in the length of *off-periods* and in the temperature differential were not of great significance.

Variations in Air Volume Delivered from High Sidewall Registers (Series 1, 5, 7-40): The points indicated by crosses and the broken line curves in Fig. 1 show the results obtained with high sidewall registers. The points shown on the four upper sets of curves indicate that no distinction could be made between the results obtained with the baseboard and the high sidewall registers. With both register locations, lower air deliveries resulted in longer time of fan operation per day, the same average number of air recirculations per hour, and in shorter lengths of *off-periods* of the fan as compared with the corresponding results for higher air deliveries.

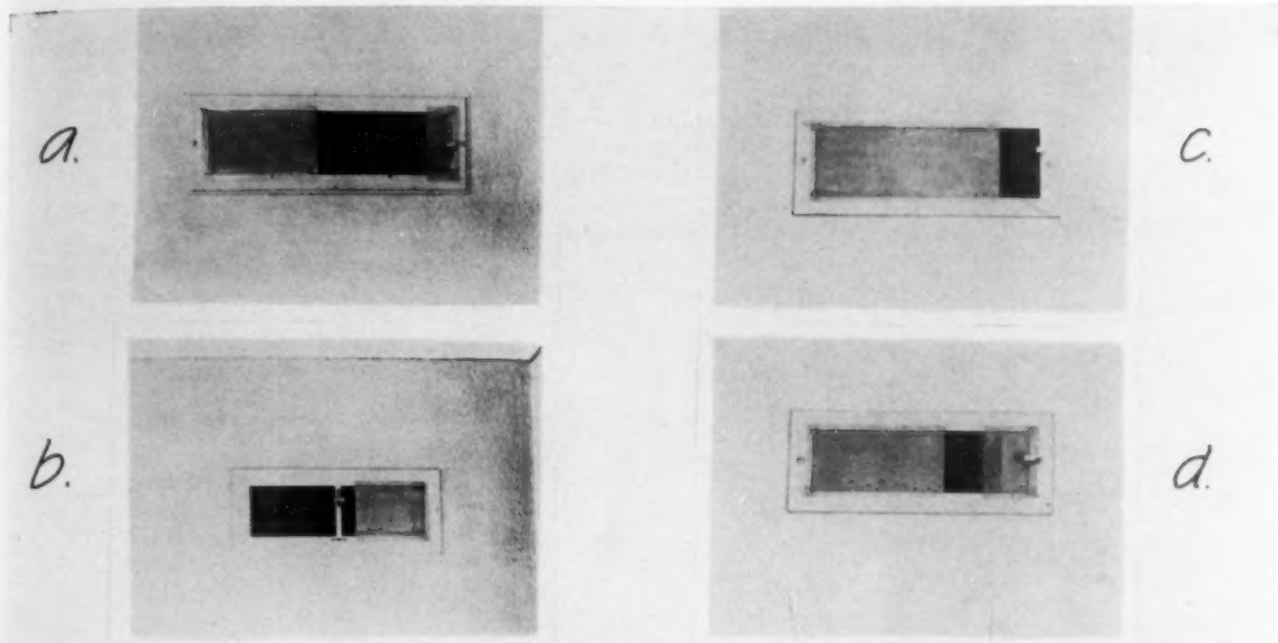
The lower set of curves indicates that with the high sidewall registers the floor temperature was not affected by the quantity of air delivered when the velocity of the air leaving the registers remained constant at approximately 600 fpm. A comparison of the floor and ceiling temperatures obtained with the two register locations shows that the floor was about 1.0 F warmer and the ceiling about 2.0 F cooler when the baseboard registers were used than they were with the high sidewall registers. The differences are not material. If large sized registers are used in connection with low air volumes, resulting in low register air velocities, higher ceiling temperatures than those shown in Fig. 1, without material changes in floor temperature might be anticipated. This is further discussed in the section on *Changes in Bonnet Air Temperature*. In this connection it is worthy of note that the use of high sidewall registers permits the employment of high register velocities and low register air temperatures without resulting drafts in the living zone, and the use of small registers that do not interfere with furniture placement.

Based on the results shown in Fig. 1 for both the baseboard and high sidewall registers it may be concluded that the number of air recirculations per hour had no immediate bearing upon either the successful operation of the plant or the comfort conditions produced when the bonnet thermostat was set to give a constant register air temperature of approximately 120 F. No data were obtained to demonstrate the result of operating the plant with a lower setting of the bonnet thermostat to obtain more nearly continuous fan operation at higher air deliveries, but under these conditions only slightly lower temperatures at the ceiling and higher temperatures at the floor level could be anticipated. The results shown in Fig. 1 further indicate that, under the usual conditions of design for intermittently operated plants and with reasonably good building construction, a register air temperature of between 110 and 140 F may be selected and the required air volume be determined from this temperature and the heat losses from the structure with reasonable assurance that the plant will operate satisfactorily. It is of course possible that in the case of poor construction in a severe climate a register air temperature greater than 140 F must be used in order not to exceed seven or eight air recirculations with the resulting difficulties from drafts.

The only feasible argument for the continued use of the arbitrary rule *that a minimum of five or six air recirculations per hour must be provided* is that in some measure it tends to compensate for the general tendency to underestimate the pressure losses in the duct system and through the furnace casing. In other words in the case of a design based on a number of air recirculations per hour as small as three, the reduced air delivery resulting from greatly underestimated pressure losses would probably be more serious than a similar reduction brought about by underestimated pressure losses in the case of a system designed with a comparatively large number of air recirculations. Rules of this nature are but temporary expedients since a deliberate overestimation of air volumes is not the best solution for an underestimation of pressure losses. The solution of the problem lies first in establishing a uniform method of rating furnace-fan units such that the air delivery of the unit is stated

Table 3—Register Opening Areas
(Actual and Calculated, for 800 cfm Air Delivery)

Room	CFM Calculated	Register Areas, Square Inches		
		Calculated	Actual	Difference
Living	144	34.6	9.6	−25.0
Hall	48	11.5	8.2	− 3.3
Dining	36	8.6	6.0	− 2.6
S. Sunroom	82	19.7	18.5	− 1.2
N. Sunroom	82	19.7	25.2	+ 5.5
Kitchen	58	13.8	18.9	+ 5.1
E. Bedroom	102	24.5	18.8	− 5.7
S. W. Bedroom	54	13.0	8.7	− 4.3
Bath	13	3.2	24.5	+21.3
N. W. Bedroom	69	16.6	22.8	+ 6.2
E. Dormitory	62	14.9	12.9	− 2.0
W. Dormitory	50	12.0	12.9	+ 0.9
Total	800	192.1	187.0	= 1.3 sq ft



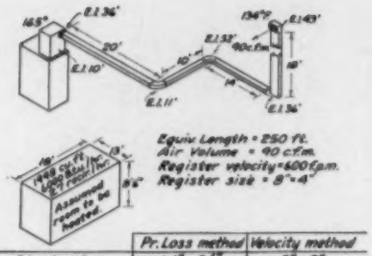
	Room	Register Size	Actual Opening	Area
a.	Kitchen	14" × 4"	3.2" × 5.9"	18.9 sq in.
b.	S. Sun	12" × 4"	3.0 × 6.1	18.5 " "
c.	Living	14" × 5"	4.1 × 2.4	9.6 " "
d.	Dining	12" × 4"	3.1 × 2.0	6.0 " "

Fig. 2—Typical register openings used with low air volume delivery

in connection with the external pressure loss of the connected duct system, and second in making accurate calculations of the pressure losses in the duct system. From the results obtained it is also apparent that the use of a large sized fan operated at slow speeds will not only provide quiet operation but will also result in short *off-periods,* and comparatively long time of operation with no increase in electrical input. Fundamentally, the success of the two-speed fan in the field has been due to the application of this principle.

Small Warm-Air Registers at High Sidewall Location: Small air volumes accompanied by high register velocities result in small sized register openings. By calculation it is evident that 800 cfm delivery accompanied by 600 fpm register velocity requires a total register opening of $\frac{800}{600}$ or 1.33 sq ft, which is equivalent to a square opening 13.9 in. on each side. Photographs of four register openings actually used in the Research Residence for the series of tests in which 800 cfm of air was circulated are shown in Fig. 2. In many cases the size of these openings was noticeably small. The calculated and actual register openings for the warm-air registers are presented in Table 3. Of the 12 registers used, 6 had actual openings of less than 15 sq in. in area, and the remainder had openings of between 18 and 25 sq in. in area.

A comparison of the calculated and actual register openings, as listed in Table 3, shows that the greatest discrepancies were obtained in the cases of the living room and bathroom. In the latter case, a considerable quantity of warm air was escaping from the bathroom into the central hall, with the result that it was necessary to supply more air than that required to heat the bathroom. In the case of the living room the actual opening was only about 28 per cent of the calculated requirement, indicating either that the heat losses were greatly overestimated or that the room was receiving heat from sources other than that supplied by the air leaving the warm air register. The latter is the most probable explanation since the room was located over the furnace and the smoke pipe. In addition, four of the studding spaces on the inside wall contained warm air stacks, and the outside wall had an adjacent warm chimney. The average temperature shown by two thermometers located in the joist spaces under the living room was about 77 F, whereas the air temperature in the center of the basement at the 60-in. level was only 66 F. For all practical purposes, the living room was heated by a panel heating system consisting of a warm floor and a warm wall. The southwest and east bedrooms were also exposed to two warm studding spaces, whereas the kitchen and sun room were ex-



	Pr. Loss method	Velocity method
Stack size	14" × 3¼"	8" × 3"
Stack velocity	280 f.p.m.	600 f.p.m.
Duct size	6" × 8"	8" × 3"
Duct velocity	270 f.p.m.	600 f.p.m.
W.A. Duct loss	0.10 in.	0.42 in.
Return duct loss	0.10 in.	0.10 in. +
Furnace press.	0.15 in.	0.15 in.
Total S.P. on fan	0.35 in.	0.67 in. +

Fig. 3—Example showing pressures in high velocity duct system

posed to relatively cool sub-floor temperatures, accounting to some extent for the discrepancies shown in Table 3.

The use of small registers in the case of well insulated structures does not necessarily imply that the stack and branch ducts can be installed with cross-sectional areas as small as those of the register openings. In the example shown in Fig. 3 it is assumed that the heat loss for a room is 6,000 Btu per hour and the connected duct has an actual length of 62 ft and an equivalent length of 250 ft. Under these conditions the register size, based on a delivery of 90 cfm and a register velocity of 600 fpm, would be 8 in. x 4 in. If it is assumed that the total loss in pressure for the entire warm air duct is to be 0.10 in. and the pressure loss method of calculation as given in the Technical Code of the *National Warm Air Heating and Air Conditioning Association* is used, the stack and duct sizes would be 14 in. x 3¼ in. and 6 in. x 8 in. respectively, both of which are practically fifty per cent greater in cross-sectional area than the 8-in. x 4-in. register.

If the velocity method of duct design is used, and a duct velocity of 600 fpm is assumed, the duct and stack sizes would be only 8 in. x 3 in., but the total loss in pressure on the warm-air side alone would be approximately 0.42 in., or 4.2 times that based on the pressure loss method. The minimum pressure loss for the entire duct system, including losses in furnace casing and return duct, would therefore be not less than, and probably greater than, 0.67 in. for the velocity method of design, and only about 0.35 in. for the pressure loss method. The fans required in the two cases would be radically different in size and performance characteristics. In fact, a fan required to deliver 800 cfm against a static pressure in excess of 0.67 in. would probably be far different from the fans now in common use for forced-air furnace units. In general, therefore, although the use of small duct systems in connection with low volume, high velocity registers is feasible, such systems must be properly designed with adequate attention given to accurate calculation of the pressure losses. Arbitrary methods of duct design such as those represented by the velocity method should not be used unless the pressure losses in the duct system are also accurately calculated, or unless fans having the necessary characteristics are provided.

Changes in Bonnet Air Temperatures (Series 4, 5, 6-40): The significant results obtained from the series of tests made with three different settings of the bonnet thermostat are shown in Fig. 4. The average bonnet air temperatures shown by the abscissae represent the mean of corresponding maximum and minimum bonnet air temperatures. All tests were made with high sidewall registers, with a total air delivery of 800 cfm, and with register openings as shown in Table 3. With the exception of one case, the register air velocity was approximately 600 fpm. The full line and broken line curves represent data for indoor-outdoor temperature differences of 34 F and 55 F respectively. The temperature differences between ceiling and breathing level, and between breathing level and floor are shown by the top set of curves. Reducing the bonnet temperature by lowering the setting of the bonnet thermostat tended to slightly lower the ceiling temperatures but had no appreciable effect on the floor temperatures. No data were obtained with different bonnet air temperatures used in connection with higher air deliveries, but Fig. 4 indicates that only slight increases in the temperature of the floor and decreases in the temperature at the ceiling could have been made by operating the plant with lower bonnet air temperatures accompanying the higher air deliveries.

One series of tests (3-40) was conducted at the low air delivery of 800 cfm for which the original size of the register was retained, thus resulting in a register air velocity of approximately 290 fpm instead of

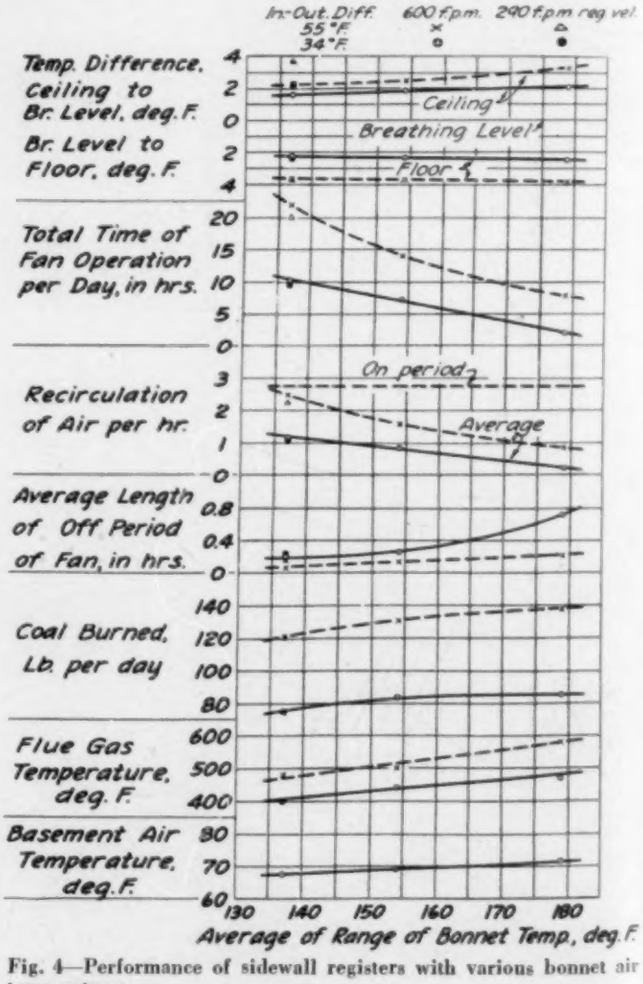

In-Out. Diff. 55 °F 34 °F
600 f.p.m.
290 f.p.m. reg. vel.
Temp. Difference, Ceiling to Br. Level, deg. F.
Br. Level to Floor, deg. F.
Ceiling
Breathing Level
Floor
Total Time of Fan Operation per Day, in hrs.
On period
Recirculation of Air per hr.
Average
Average Length of Off Period of Fan, in hrs.
Coal Burned, Lb. per day
Flue Gas Temperature, deg. F.
Basement Air Temperature, deg. F.
Average of Range of Bonnet Temp., deg. F.

Fig. 4—Performance of sidewall registers with various bonnet air temperatures
800 cfm delivery, stoker-fired furnace

600 fpm. The results are shown by the triangles and solid circles in Fig. 4. The only noticeable effect with the comparatively low bonnet thermostat setting was an increase in the temperature of the air at the ceiling.

Although the effect of reduced setting of the bonnet thermostat on the room temperature gradients was small, in other respects the most satisfactory operation was obtained with the lower values of the bonnet air temperature.

As shown by Fig. 4, reducing the bonnet temperature resulted in a longer total time of fan operation per day, a larger average air recirculation, and much shorter *off-periods*. In fact the setting of the bonnet thermostat can be so reduced as to produce practically continuous fan operation even on days in which the outdoor temperature is not extreme. With such reduced settings of the upper limit of the bonnet thermostat, above which temperature the fan would operate, if the limit thermostat for the stoker or burner were set at a temperature high enough to take care of extreme weather conditions, the operating bonnet air temperature would be automatically proportioned to the heating demands of the structure. That is, in very cold weather the operating bonnet air temperature would automatically rise higher than the temperature indicated by the upper limit of the bonnet thermostat setting, with the net result that the heat output would be adequate. On the other hand, if the bonnet thermostat setting had been increased to a very high value, the plant would have tended to operate entirely under gravity air circulation. With a high setting of the bonnet thermostat and a given adjustment of the duct dampers the temperature balance between rooms over a wide range of outdoor temperatures was not as satisfactory as that obtained with a similar adjustment in the case of lower settings of the bonnet thermostat. Higher bonnet settings, particularly for a solid fuel plant in which some heat liberation occurs during the *off-periods* of the stoker, result to some extent in uncontrolled gravity circulation of the air during the *off-periods* of the fan and thus tend to produce inequalities in the temperature distribution between rooms.

As shown by Fig. 4, the higher settings of the bonnet thermostat were also accompanied by greater coal consumption on a given day. The increase in coal consumption could be largely accounted for by an increase in the heat losses from the basement resulting from higher basement temperature and by an increase in the flue losses accompanying higher flue gas temperatures. The latter was particularly true during the *off-periods* of the stoker.

From the standpoint of low temperature difference from floor to ceiling, long total time of operation of the fan, shorter lengths of *off-periods*, and lower fuel consumption, the most favorable results were obtained with the lower settings of the bonnet thermostat. In this connection it should be noted that when lower bonnet air temperatures are used the actual operation of the plant more nearly approaches that with continuous operation of the fan, thus approximating the conditions assumed for design. Furthermore, the amount of dampering required in the duct system to take care of variations in outdoor temperature will be dependent upon the range of bonnet air temperatures maintained, and should be the least with low ranges of bonnet air temperatures. Lower bonnet air temperatures can be used in conjunction with high sidewall or ceiling registers than with baseboard registers. The lowest allowable limit of the bonnet air temperature is dependent upon the lowest register air temperature tolerable without resulting in objectionable drafts in the rooms.

Summary

The results of these tests may be summarized as follows:

1. The forced-air heating system proved to be very flexible insofar as its adequacy to heat the Residence with air volumes and register air temperatures deviating from those selected for the design of the plant was concerned.

2. At a bonnet air temperature of approximately 150 F and with a register air velocity of approximately 600 fpm, the variations in the fan delivery equivalent to from three to six air recirculations per hour had no immediate bearing upon either the successful operation of the plant or the comfort conditions produced.

3. With intermittent operation of the heating plant at an approximately constant bonnet air temperature, although the fan delivery varied from three to six air recirculations per hour, the average air recirculations per hour including both the *on-periods* and *off-periods* remained constant.

4. The use of baseboard registers resulted in floor temperatures approximately 1.0 F warmer and ceiling temperatures approximately 2.0 F cooler than those obtained with high sidewall registers. The use of high sidewall registers permitted the employment of high register velocities and low register air temperatures without resulting drafts in the living zone.

5. The use of high sidewall registers in connection with low air volumes and high register air velocities permitted the employment of registers smaller than those usually observed in common practice.

6. The use of an excessively small duct system in connection with small registers cannot be justified unless accurate calculations of the anticipated pressure losses are made.

7. From the standpoint of low temperature difference between floor to ceiling, long total time of operation of the fan, shorter lengths of *off-periods*, and lower fuel consumption, the most favorable results were obtained with the lower settings of the bonnet thermostat.

Acknowledgment

The results presented in this paper were obtained in connection with the investigation of warm-air furnaces and heating systems in the Research Residence[3] at the University of Illinois, conducted by the Engineering Experiment Station. This investigation is a cooperative project sponsored jointly by the *National Warm Air Heating and Air Conditioning Association* and the Engineering Experiment Station. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is made to D. W. Thomson and H. R. Stover, formerly Research Graduate Assistants, for their services in conducting the tests and in the reduction of the test data. Acknowledgment is also made to the various manufacturers who cooperated by furnishing instruments and equipment.

[3]The Research Residence in Urbana, Illinois, was built, furnished, and completely equipped specifically for research work in warm-air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

Measurement of the Physical Properties of the Thermal Environment

Report of the ASHVE Research Technical Advisory Committee on Instruments: D. W. Nelson, *Chairman*; F. R. Bichowsky, L. M. K. Boelter, R. S. Dill, A. P. Gagge, John A. Goff, A. E. Hershey, F. C. McIntosh, F. W. Reichelderfer, G. L. Tuve, C. P. Yaglou.

Introduction

RECOGNIZING THE need for greater uniformity in reporting the measurement of the physical properties of a thermal environment, this committee has prepared this tentative outline of testing procedure for the items involved. The four basic physical properties of the thermal environment are: relative humidity, ambient air temperature, air movement and mean radiant temperature.

This outline is intended for the guidance of research laboratory workers in this field and is not to be construed as a standard with which the industry is expected to comply. It is an outline for assistance and not a standard for enforcement. In the discussion, generally a definition will be given in terms of an ideal method of measurement, followed by practical means of measurement. As detailed knowledge is gained, additions and changes are in order. The committee wishes to encourage the development of basic measurements on the part of investigators who are working on fundamental research in this field and it does not wish to standardize on any definite procedure at the present time.

Relative Humidity

There are various ways of measuring the water vapor present in atmospheric air. A common and accurate method is to measure the wet- and dry-bulb temperatures by a sling psychrometer or an aspirating psychrometer. The velocity past the wet-bulb wick should be about 1000 fpm. Determination of relative humidity and other properties may be obtained by use of a formula or by chart. The latter usually applies to only one barometric pressure, whereas the formula usually allows for variations in pressure.

The amount of water vapor may be determined by measuring the dew-point temperature, which may be done by observing the appearance of a slight fog on a highly polished surface whose temperature is also measured. The lowering of temperature may be secured by vaporizing a volatile fluid or by the expansion of a vapor such as carbon dioxide. Care must be taken to obtain the surface temperature where the vapor condenses, rather than that of the fluid. The conductivity over a glass surface between electrodes has also been used. When moisture condenses, the conductivity greatly increases. The increase may be read on a suitable instrument with simultaneous measurement of the glass surface temperature.

Moisture may be removed by chemical drying and the amount determined by analytical weighing. This method requires considerable apparatus and care, so is not recommended for general use.

The hygroscopic properties of material may be used for moisture determination. The change in length of hair with variations in moisture content of atmospheric air is often used. This method is not to be recommended for careful work, without frequent calibrations at various moisture contents and ambient temperatures.

Ambient Air Temperature

Definition: The ambient air temperature of an environment is the equilibrium temperature reached by an ideal thermometer with zero radiation emissivity *i.e.*, with perfect reflectivity, and of negligible thermal capacity in order to reach equilibrium in a short time.

In practice, all thermometric measuring devices, whether thermocouples, mercury or gas thermometers, are not perfect reflectors and have various degrees of thermal capacity. In any practical thermometric procedure, either the relative thermal capacity of the measuring device to the environment is reduced or the relative temperature response to radiation is reduced in relation to its response to convected heat, or both. Locations of thermometers should be stated, and one at the 30 in. level should be included. Where gradients are important it is suggested that observations be made at a series of levels. The methods generally used to measure ambient air temperature and precautions to be observed are listed herewith:

1. *Aspirating Thermometer:* The general design and requirements are:

a. As much as possible of the aspirated air should pass over the sensitive bulb of the thermometer without introducing expansive cooling of the air itself.

b. The bore and bulb size of the thermometer should be so chosen that quick response to the air stream is possible.

c. The bulb of the thermometer should be reasonably well shielded from any outside radiation.

A certain length of time is required to reach equilibrium depending upon design factors such as, the rate of aspiration and size of bulb and the original difference between the temperature of the instrument and of the air. Sufficient time must be allowed to reach equilibrium, generally noted by taking successive readings with the difference between one and the next approaching zero. Experience will indicate a certain time interval to allow between observing near equilibrium and the observing of the final reading.

2. *Aspirating Thermocouple:* With a thermocouple in place of the glass thermometer:

a. A smaller volume of aspirated air is necessary than with a thermometer since the thermal capacity of a couple is smaller.

b. No particular precautions are necessary in the design and size of a couple when proper shielding from direct radiation is possible. However, fine bright wire with a small junction surrounded by

a polished metal shield is not difficult to use and is on the side of safety.

3. *Shielded Thermometer:*

a. The shield should be a polished metal cylinder open at both ends for free circulation; copper and aluminum are satisfactory.

b. A height of cylinder equal to its diameter is usually satisfactory. The diameter should not be made too small so as to interfere with free air circulation.

c. The sensitive part of the thermometer should not *see* radiant heat sources. In some cases a longer shield is justifiable to eliminate radiant effects and at the same time the diameter might be increased so as not to interfere with air circulation. Another expedient would be to place the axis of the thermometer other than in a vertical position where locations of radiant sources make this advisable.

d. In locations near intense radiant sources multiple shields of concentric cylinders might be necessary to eliminate radiant effect.

e. The thermometer should be selected for the immediate range to be measured.

4. *Shielded Thermocouple:* In addition to the items previously mentioned, when a thermocouple is used, the time response may be expedited by making it of small wire; down to as small as No. 36 has been found satisfactory. The use of small wire is in the direction of guarding against radiant effects.

5. *Free Thermocouple:* For locations where radiant sources are neither near nor too strong, a free thermocouple gives reliable ambient air temperatures when properly made.

a. The thermocouple wire should not be larger than No. 30 and might well be as small as No. 36.

b. Silver solder or soft solder may be used for the junction, although the former tends to make an enlarged bulb which increases radiant effect. Welding, when skillfully done gives a smaller junction. This can be done in a carbon arc or with low voltage current and an oil covered mercury bath.

c. The junction should be cleaned and polished. Copper constantan acquires and maintains a good polish and is satisfactory in other ways.

d. At times the junction is flattened into a thin ribbon and trimmed to reduce thermal capacity and increase rate of response by increased exposed area in relation to its volume. This is usually not advisable when the small wire sizes are used. The response of the small wire is fast and the relationship between radiation and convection pick-up is favorable to the cylindrical wire. As wire size decreases the convection effect increases and the radiant effect decreases, so it is possible to have a small well-made free thermocouple give a more accurate ambient temperature observation than does a large dull surfaced thermocouple in a shield of poor design or construction. Where slow response is desired to obtain a mean value a thermocouple may be buried in metal, conveniently of spherical or spheroid shape with a highly polished surface, but would be subject to error if radiant sources are near. In all thermocouple work it is necessary to have indicating or recording instruments that permit the accuracy required.

6. *Sling Thermometer:* This type is similar to the aspirating thermometer since it increases convection by increased air circulation. The nearness of the person using it may cause radiation effects especially in the process of stopping to read. Oftentimes the sling thermometer is used in conjunction with a wet-bulb thermometer, in which case care must be taken to keep moisture from the dry-bulb.

Each of the described methods for obtaining ambient air temperature has merits and limitations. These must be kept in mind in relation to the particular location involved.

Air Movement

The air movement mentioned in this report is that associated with confined spaces for human occupancy. In general, the velocities do not exceed 150 fpm in regions of occupancy and usually range from 100 fpm down toward a minimum of about 10 to 25 fpm. The higher velocities usually exist in rooms when air is supplied to the room by mechanical means and the lower velocities where convection currents are established by heating or cooling units or surfaces in the room.

The motion of such air masses is largely non-directional. Although the main transport may be in a certain direction, any one small part of the air mass is travelling in a continuously varying path. For purposes of defining the thermal environment, this non-directional, rather than uni-directional movement, is the important item in velocity.

Some of the instruments for measuring low velocity non-linear air movements are:

1. *The Kata Thermometer:* This anemometer uses the time in seconds to cool a liquid in glass between two fixed temperatures. It is made in two ranges, termed the low temperature for temperatures below about 85 F and the high temperature for higher ambient temperatures. Cooling of the liquid takes place by radiation and convection currents due to the instrument being warmer than its surroundings and by air movement from external sources. In some cases the bulb is silvered to reduce the radiation effect, which calls for a different calibration from that for the plain bulb. Due to the disturbance of convection currents, all of the cooling by air movement is not that of the air current to be measured. The instrument is affected by direction of the air movement, as for instance, near a wall where the movement may be upward or downward. With upward movement the convection current is additive and with downward movement this current is subtractive; consequently the Kata would show higher than the true velocity in the upward current and less with the downward current. As usually made, the bulb is large, which is in the direction of more disturbance when used near radiant surfaces. When the difference between the ambient temperature and mean temperature of the Kata is greatest, the convection effect is the greatest. For exactness, individual calibrations should be available to take care of various radiant energy situations, various mean temperature differences and general drifts of air mass such as vertically upward, downward or horizontal. The rate of response is slow, especially at low temperature differences and low velocities. The determination is a mean value for the time involved which may be from a few seconds to many minutes. The instrument finds considerable use and has a lower limit under certain conditions of about 25 fpm.

2. *The Heated Liquid in Glass Thermometer:* The difference in temperature between the temperature of a mercury thermometer with a bulb heated by electrical resistance and a thermometer with a plain bulb is used to measure air movement. By using a low temperature difference, the disturbance of convection currents is kept at a low level and the instrument is fairly omnidirectional. Because of the mass of material involved in the sensitive element, the response is slow and the

reading is a mean over a considerable period of time. With proper selection of mass and temperature difference, the range of air movements usually encountered in thermal environments can be covered.

3. *The Hot-Wire Anemometer:* This classification is intended to include the thermocouple anemometer as well as the hot-wire anemometer. The principle is to pass a current through a small resistance wire in free space and to measure the effect of air movement on its temperature by change in resistance with a potentiometer or with a thermocouple. There are many variations of this type, each with its own characteristics as to such items as rate of response, range with good accuracy and sensitiveness to directional air movements. A single straight wire is most sensitive normal to its length, so for non-directional measurement three mutually perpendicular wires are often used. By making the heated wire very small in diameter, the time response can be made very rapid so that minute fluctuations can be observed. However, a slower rate of response is desirable so that a mean value is obtained directly and this slower response may be secured by using a larger size of wire. A reasonable rate of response would appear to give a mean reading in the range of one to ten seconds.

All of these instruments for measuring air movement operate on the basic principle of the cooling of a heated element in the air stream. Choice of an instrument should depend on such items as, the determination of a mean value which is associated with a reasonable speed of response, ease and accuracy of calibration, ease of manipulation and non-directional response.

Calibration of Anemometer

For sake of agreement a common method of calibration should be followed, for it is essential that the method of calibration be standardized in order that test results be comparable. Considerable work has been done on the development of wind tunnels for low velocity instrument calibration. Essentially the arrangement consists of an entrance cone followed by a transparent duct section in which the instrument element is placed. From this section the air passes through a duct section with a rounded-approach nozzle for the purpose of flow measurement. Motive power for drawing the air through the tunnel or duct is by an electrically driven fan. The purpose of the entrance cone is to insure uniform flow across the instrument section. The nozzle must be made to careful standards for which the coefficients of discharge at various volumes are known or else receive accurate calibration. The duct and flow nozzle size must be chosen with regard to the velocity range for which the instrument is to be calibrated. The fan must have a capacity to draw through the required amount of air against the resistance of the tunnel system, of which the drop through the nozzle will likely be a large part. The wind tunnel appears to have the most promise for low-velocity meter calibration.

As experience is gained, perhaps other methods of calibrating will be proven to be equally reliable. The whirling arm may prove to be adaptable to this use and the displacement gas tank or gasometer has possibilities. Air in free space has a non-linear motion that varies widely, depending upon the forces acting upon it. The use of the wind tunnel is an attempt to suppress non-linear motion and make linear motion predominant. Various types of instruments have different rates of response, and thus are affected differently by non-linear air currents. For this reason it is not to be expected that a Kata-thermometer anemometer and a hot-wire anemometer, properly calibrated in the wind tunnel, will give equal readings under turbulent conditions as encountered in free space under heating or cooling conditions. For complete uniformity of results the type of instrument, the method of calibration, and the exact details such as the type of hot-wire anemometer and the size and arrangement of elements should be specified.

Mean Radiant Temperature

Definition: The mean radiant temperature of an environment is the temperature of a uniform enclosure with which one's body would exchange the same amount of energy by radiation as in the actual environment.

This definition, although adequate, does not convey fully the real significance of mean radiant temperature which may be deduced from the Stefan Radiation Law in its most general form. For two surfaces at temperature T_1 and T_2 F absolute, the heat exchange is given by:

$$R = 0.172\,A\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right]F_A F_E \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

where

R = the net rate of heat exchange, Btu per hour.

A = area of one of the two surfaces, square feet.

F_A = a factor allowing for the average angle with which one surface *sees* the other and is obviously dependent on which surface is used in the term A.

F_E = a factor allowing for departure of the two surfaces from complete blackness and is a function of P_1 and P_2, the respective emissivity of the two surfaces.

For purposes of analysis, if one assumes the object receiving or emitting radiation is either a man, thermo-integrator, globe thermometer, or any solid object with uniform surface temperature, T_g, and emissivity, P, and if this object is losing radiation to an enclosure with black body walls but with two different surface temperatures, T_W' and T_W'' with subtended angle factors F_A' and F_A'' at the receiving object, Equation (1) reduces to:

$$R = 0.172 \times 10^{-8}\,AP[F_{A'}\,(T_g^4 - T_{W'}^4) + F_{A''}\,(T_g^4 - T_{W''}^4)] \quad \ldots\ldots\ldots\ldots (2)$$

But by definition for mean radiant temperature, T_W,

$$R = 0.172 \times 10^{-8}\,A \times P(T_g^4 - T_W^4)\,. \quad (3)$$

Comparing (2) and (3):

$$F_{A'} + F_{A''} = \text{unity}, \quad \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

and

$$T_W^4 = F_{A'}T_{W'}^4 + F_{A''}T_{W''}^4 \quad \ldots\ldots\ldots\ldots\ldots (5)$$

In this example, the mean radiant temperature is an average of the temperatures of the radiating surfaces weighted according to the angle they subtend at the receiving object. Thus, mean radiant temperature applies only to the object receiving the radiation and is a function (through F_A' and F_A'') of both the shape of the receiving object as well as the geometrical position of the radiating surfaces. Therefore, in actual practice, it is important to note that a man, a thermo-integrator, or a globe thermometer *see* numerically different radiant wall temperatures.

The thermopile is the only practical physical instrument that can measure the effect of radiation without regard to ambient air temperature or air movement. The thermopile may be used to measure only the temperature of a surface it sees. For an object of any shape receiving radiation, one could sample the temperature it sees normal to its surface uniformly all over the total surface of the object. A proper average of these temperatures would be the true mean radiant temperature for that object. Such a procedure, although laborious, offers the only method available by which one can measure directly the mean radiant temperature as seen by an object. Thus, the fact remains that the sampling thermopile technique is the fundamental method of measuring the mean radiant temperature of any object.

If one knows quantitatively how an object loses heat by convection, C, to its environment and if one can measure the mean surface temperature of the object itself, it is possible to deduce the mean radiant temperature for that object from Equation 3 since the heat gain or loss by radiation equals the heat loss or gain by convection. Further, if the object generates heat, H, then, of course, R, is the difference between H and C.

The three well-known radiation-convection integrating devices, eupatheoscope, thermo-integrator, and globe thermometer, use this general principle. Each consists of a metallic shell without negative curvature in which there may or may not be a heat source. By operation under known radiant conditions, convection properties of each instrument may be quantitatively determined. With the additional knowledge of the reflectivity factor for the surface of the receiver to the type of radiation being measured, any of these instruments may be used to measure mean radiant temperature, which, of course, would be valid by definition only for the individual instrument. The general applicability of these instruments to thermal environment for human comfort would be based on the knowledge of a factor, above or below unity, which relates radiation exchange per unit area of each instrument to a corresponding exchange for the human body under like circumstances.

Of the three types of instruments mentioned it is recommended that the globe thermometer be the standard field instrument for the measurement of mean radiant temperature. Recommendations for the construction, use and calibration of the globe thermometer are outlined herewith:

1. *Design of the globe thermometer:*

a. The master globe thermometer is made from an 8 in. O.D. copper sphere having a wall thickness of 0.022 in. The use of smaller diameters is not advised, although correction curves are available to change results to the 8 in. equivalent. A spud with ¼ in. IPS internal thread is soldered to the sphere for support.

b. The temperature of the globe may be measured by an accurate mercurial thermometer, calibrated from 0 to 120 F; a 12 in. length of ¼ in. diameter is convenient. The thermometer should be held in position by a rubber bushing in the spud so that the bulb of the thermometer lies at the center of the globe. Response may be quickened by blackening the thermometer bulb without affecting accuracy. A thermocouple may be used in place of the glass thermometer. For this purpose the globe should be supported by a 4 in. long fibre rod, ½ in. in diameter. One end of the rod should have a ¼ in. IPS thread. The couple is inserted in the bore of the rod. Where thermocouples of No. 34 wire are used for ambient air temperatures, it is convenient and advantageous to use the same type for obtaining the globe temperature.

c. The outside surface of the globe shall be blackened with two coats of Flat Black—Sherwin Williams paint No. 42. The emissivity of the globe so blackened has been determined to be 94 per cent whether paint is applied by brush or spray gun.

d. The globe should be supported by light twine connected to the spud or by some other means leaving the surface free to radiation.

2. *Use of the globe:*

a. The location of the globe in the room, including the distance from the floor and from the adjacent walls, should be recorded. In ordinary environments it is suggested that the mean height of the globe be 30 in. above the floor, although other heights may be investigated also.

b. The air movement shall be measured at the globe location by an anemometer properly selected and calibrated.

c. The ambient air temperature may be taken with an aspirated thermometer at a distance of 1 ft from the side of the globe. In ordinary environments where the quantity of radiation present is such that the difference between the temperature of the air and globe is less than 10 F, the ambient air temperature may be closely determined with one or more thermocouples made of wire not larger than No. 30 and located 2 in. from the surface of the sphere. This couple or couples, should be located in a horizontal plane passing through the center of the globe. Because of the inherent sluggishness of the globe, its reading represents an average for quite a long period of observation, and the observed air temperature should represent an average over the same period. The use of thermocouples and electrically heated anemometers makes it possible to take readings remotely, thus removing any effect of the observing body.

d. The mean radiant temperature (T_w) determinations are made by using a modified Bedford formula and the observed ambient air temperature (T_A), air movement (V), and globe temperature (T_G). This formula is:

$$0.94 \times 1.72 \times 10^{-9} (T_w^4 - T_G^4) = 0.164 \sqrt{V} (T_G - T_A) \quad \ldots\ldots\ldots\ldots\ldots (6)$$

e. When globes of similar construction but of smaller diameters are used, the coefficient for convection loss in the right hand term of the Bedford equation must be altered. The values of this coefficient for different globe diameters are:

Diameter of Globe	Convection Coefficient
8 in.	0.164
6 in.	0.169
4 in.	0.181
3 in.	0.188
2 in.	0.202

Changes in the convection coefficient for globes above 8 in. diameter are believed to be negligible. The 6 in. globe was used by Bedford as a standard.

f. When taking measurements with the globe thermometer, whether for calibration purposes or in practice, it is important that all couples and thermometers used be calibrated against a master thermometer over the temperature range involved.

Summary

An effort should be made to standardize measurements of the physical properties of the environment which will result in greater uniformity and continuity of data in relation to the findings of various investigators. The measurements of temperature, whether radiant or ambient air, and of air movement, are part of a language used to describe the thermal environment. The obvious and natural goal is to strive for standardization in measurement in order to describe the environment in terms of a common language.

A Method of Compiling Tables for Intermittent Heating

By Elmer G. Smith*, College Station, Tex.

Introduction

THE PROBLEM of accurately determining loads for intermittently heated buildings is one that has concerned engineers who have specialized in churches. A number of formulae have been developed for calculating the heating requirements of such buildings and often inadequate plants are installed. For this reason many churches require long heating-up periods and in some cases improper comfort conditions result.

One factor that is frequently overlooked in designing systems for intermittently heated buildings is the specific heat of the building materials. This factor is frequently referred to as storage effect or lag, and in massive structures this is an important factor.

The solution of this problem has demanded a thorough investigation of the transfer of heat through building materials. Up to the present time practical information concerning the absorption of heat in a building material has been lacking. From the theoretical derivations outlined in this paper, it is hoped that the engineer and contractor will ultimately have theoretically sound formulae which may be used with a certain degree of confidence. It has been possible to develop constants giving the heat absorbed in Btu per hour per square foot of surface, per degree rise in air temperature for different floor and wall constructions. This value, added to the customary heat transfer load through a building wall, would make it possible to determine the necessary *additional capacity* for an intermittently heated building.

Previous papers[1] describe the derivations of equations which give the rates at which simple (single layer) walls absorb heat under the conditions that usually exist in intermittently heated buildings. These equations are based upon the following assumptions: (1) Previous to the starting of the heating system, the air in a given room and all the walls are at the same temperature. (2) At the instant the heating plant is started, heat suddenly begins to enter each wall. (3) The rate at which heat enters any given wall is constant until the building has become comfortable. (4) The outdoor temperature remains constant.

It should be noted that it is not assumed that the temperature of the air in the building is suddenly raised to 70 F and then remains at that temperature until the walls have reached their steady state temperatures. This would give results much too high. Under the assumptions actually made, the temperature of the air suddenly rises, but not to 70 F. After the first sudden rise the temperature difference between the air and the wall surface remains constant, while the air temperature is being increased to 70 F.

In Bulletin[2] No. 60 of the Texas Engineering Experiment Station is a set of tables based upon the foregoing equations and applicable to either intermittent heating or intermittent cooling. Although the process of computing them is laborious, the tables are no more difficult to use than the tables of conductances for various walls that are given in THE GUIDE. The numbers which make up the main body of the tables have been called the absorbances of the walls because of their similarity to the conductances which are used in the familiar steady state calculations. Table 1 shows the parts of them that refer to monolithic exterior walls of sand and gravel concrete, and to brick interior walls. To the left of the double line are given the constants used in calculating the absorbances of the walls listed.

The use of the tables can best be explained by a numerical example. Suppose a given building to have 9,000 sq ft of exterior wall made of monolithic concrete 10 in. thick. The temperature of the interior of the building is never to be allowed to fall below 40 F, but the outdoor temperature may drop to 0 F. Thus the losses between 40 F and 0 F are steady state, while those between 70 F and 40 F are transient. Also, suppose that the building is to be comfortably warm 3½ hours after the fires are lighted. Since approximately one-half hour will probably be needed to raise the temperature of the heating plant sufficiently to enable it to deliver heat at an appreciable rate, the time required to heat the building must be taken as 3 hours. In the upper portion of Table 1, at the intersection of the column headed by 3 hours and the row beginning 10 in. of concrete, the figure 1.001 may be found. Consequently the transient loss for the 9,000 sq ft of wall may be determined by multiplying the wall area by the absorbance and temperature rise of the air.

9,000 (1.001) (70-40) = 270,300 Btu per hour.

Similarly by reference to THE ASHVE GUIDE[3] the steady state loss is found to be:

9,000 (0.62) (40-0) = 223,200 Btu per hour.

The total heat requirement for the wall, therefore, is the sum of the steady state and transient losses.

270,300 + 223,200 = 493,500 Btu per hour.

Suppose that the interior of the same building contains 12,000 sq ft (counting both sides) of 4-in. brick partitions plastered on both sides. In the lower portion of Table 1, at the intersection of the column headed by 3 hours and the row begun by 4-in. brick, the figure 0.720 may be found. Therefore, the

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[1]A Simple and Rigorous Method for the Determination of the Heat Requirements of Simple Intermittently Heated Exterior Walls, by Elmer G. Smith. (*Journal of Applied Physics*, Vol. 12, No. 8, August, 1941, p. 638.)

The Heat Requirements of Simple Intermittently Heated Interior Walls and Furniture, by Elmer G. Smith. (*Journal of Applied Physics*, Vol. 12, No. 8, August, 1941, p. 642.)

[2]Heat Requirement Tables for Intermittently Heated Buildings. (*Engineering Experiment Station Bulletin*, No. 60. A. and M. College of Texas, College Station, Tex.)

[3]HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1940, p. 100.

Table 1

ABSORBANCES OF EXTERIOR WALLS IN BTU PER SQUARE FOOT PER HOUR PER DEGREE FAHRENHEIT

TYPE OF WALL	THICKNESS OF WALL IN INCHES	PER SQUARE FOOT OF INTERIOR SURFACE				TIME IN HOURS								
		INTERIOR SURFACE RESISTANCE	EXTERIOR SURFACE RESISTANCE	RESISTANCE OF WALL	THERMAL CAPACITANCE	½	1	2	3	4	5	6	7	8
Concrete (Monolithic Sand and Gravel)	6	0.63	0.167	0.48	13.5	1.291	1.192	1.075	1.002	0.949	0.911	0.883	0.861	0.844
	10	0.63	0.167	0.8	22.45	1.290	1.191	1.074	1.001	0.949	0.906	0.868	0.839	0.815
	16 or more	0.63	0.167	1.28	36.0	1.289	1.190	1.072	1.000	0.948	0.904	0.866	0.838	0.814

ABSORBANCES OF INTERIOR WALLS IN BTU PER SQUARE FOOT PER HOUR PER DEGREE FAHRENHEIT

TYPE OF WALL	THICKNESS OF WALL IN INCHES	PER SQUARE FOOT OF INTERIOR SURFACE			TIME IN HOURS								
		INTERIOR SURFACE RESISTANCE	RESISTANCE OF WALL	THERMAL CAPACITANCE	½	1	2	3	4	5	6	7	8
Brick (½ in. of Plaster on each side)	4	0.63	0.475	4.99	1.141	1.021	0.842	0.720	0.629	0.559	0.502	0.456	0.418
	8	0.63	0.9	9.45	1.139	1.021	0.889	0.814	0.738	0.685	0.639	0.597	0.563
	12	0.63	1.325	13.9	1.141	1.021	0.890	0.810	0.754	0.706	0.670	0.637	0.608
	16 or more	0.63	1.75	18.4	1.141	1.021	0.891	0.810	0.755	0.710	0.674	0.644	0.619

	TIME IN HOURS AS ABOVE								
Infiltration and Glass Factor $\frac{T+2}{T+3}$	0.71	0.75	0.80	0.83	0.86	0.88	0.89	0.90	0.91

T = Time in hours.

transient loss for 12,000 sq ft (6,000 sq ft if the surface on only one side of each partition is counted) is found to be:

12,000 (0.720) (70-40) = 259,200 Btu per hour.

Since there is no steady state loss for an interior wall, 259,200 Btu per hour is the total heat requirement for these partitions.

At the bottom of Table 1 is a line of figures designated as Infiltration and Glass Factor. These numbers resulted from a conservative estimate as to how much the infiltration and glass losses will be reduced on account of the fact that the interior of the building is at a low average temperature during the transient period. Thus, in computing the infiltration and glass losses for the previously described building, instead of using 70 F — 0 F = 70 F, one should use (40 F — 0 F) + 0.83(70 F — 40 F) = 64.9 F, where 0.83 is the infiltration and glass factor for a 3-hour transient period.

The heat requirements of roofs and very heavy outside doors are obtained in the same manner as for the exterior walls. Thin panels in outside doors should be treated in the same manner as glass. Inside floors and furniture should be treated in the same manner as interior walls. Unless the building is very small, the heat required to raise the temperature of the air should also be considered.

Before leaving the subject of previous work it should be mentioned that in writing the bulletin on intermittent heating, every effort was made to use conservative values in computing the tables. In accordance with this policy, the surface conductance used for intermittently heated interior surfaces was the same that is usually used for continuously heated interior surfaces. Since all walls receive much less radiation during the transient period when the interior as well as the exterior walls are cold, the resulting absorbances are too high. The bulletin gives a simple method of converting an absorbance based on one surface resistance to that based on another so that the variation can be remedied as soon as the proper values have been definitely ascertained.

Purpose

Since new building materials and new types of structures continually make their appearance, it is desirable that anyone interested be able to compute the absorbance of a wall without being obliged to read through the original derivations in order to be certain that the process is correct. Accordingly the purpose of this paper is to give detailed illustrations showing exactly how each of the equations is to be used to obtain new values of absorbances.

Equation for Simple Interior Walls

It is hoped that the procedures and examples which follow are sufficiently clear to enable engineers to compute the absorbances of new types of walls whenever that seems desirable, either because a new type of wall has come into common use or because the physical constants of some wall already in use have been more carefully determined.

For a simple interior wall, it can be proved that, if heat is supplied to a wall at a constant rate, its absorbance is given by the equation[4]

$$a = \frac{1}{R_s + \frac{R}{3} + \frac{T}{C} - \sum_{n=1}^{n=+\infty} \frac{\left(\frac{2R}{\pi^2}\right)}{n^2 e^{\left(\frac{\pi^2}{RC}\right) T n^2}}}$$

where

$a \equiv$ absorbance of the wall.

$R_s \equiv$ surface resistance of the wall per square foot.

$R \equiv$ resistance of the wall between one of its surfaces and its mid-plane for each square foot of surface.

$T \equiv$ time that may be allowed to elapse between the time when appreciable heat begins to enter the auditorium or room and the time when it first becomes comfortable.

$C \equiv$ thermal capacitance of the wall between its surface and its mid-plane per square foot of surface.

$e \equiv$ base of Naperian logarithms.

$n \equiv$ any positive integer.

[4]Loc. Cit. Note 1 (second reference).

Procedure for Simple Interior Walls

1. Calculate C = (density) (specific heat) (half the thickness of the wall expressed in feet).
2. Record R_s
3. Record R
4. Calculate $\frac{2R}{\pi^2}$
5. Calculate $\frac{\pi^2}{RC}$
6. Calculate as many terms of the summation as are needed.
7. Substitute in the equation to obtain the absorbance.

Examples

For a 2-in. wall of solid plaster in a room that is to be brought to a comfortable temperature in 3 hours:

1. $C = 110(0.20)\left(\frac{1}{12}\right) = 1.833$
2. $R_s = 0.63$
3. $R = 0.30(1) = 0.30$
4. $\frac{2R}{\pi^2} = \frac{2(0.30)}{\pi^2} = 0.0608$
5. $\frac{\pi^2}{RC} = \frac{\pi^2}{0.30(1.833)} = 17.95$

$$6.\ \sum \frac{\left(\frac{2R}{\pi^2}\right)}{n^2 e^{\left(\frac{\pi^2}{RC}\right)\tau n^2}} = \sum \frac{0.0608}{n^2 e^{(17.95)3n^2}} = \frac{0.0608}{(1)^2 e^{53.85}} + \frac{0.0608}{(2)^2 e^{107.7}} + \cdots \cong 0$$

7. $$a = \frac{1}{0.63 + \frac{0.30}{3} + \frac{3}{1.833} - 0}$$
$$= \frac{1}{0.63 + 0.10 + 1.636}$$
$$= \frac{1}{2.366} = 0.422 \text{ Btu per hour per square foot per degree Fahrenheit}$$

The summation is negligible for the above wall and also for most interior walls in the types of buildings being constructed at the present time because of their thinness. As an example of the other extreme, take the case of a 24-in. brick wall in a building that is to be made comfortable in 3 hours. In this case the various steps are:

1. $C = 120(0.21)1 = 25.2$
2. $R_s = 0.63$
3. $R = 0.20(12) = 2.40$
4. $\frac{2R}{\pi^2} = \frac{2(2.4)}{\pi^2} = 0.4863$
5. $\frac{\pi^2}{RC} = \frac{\pi^2}{2.4(25.2)} = 0.1632$

$$6.\ \sum \frac{\left(\frac{2R}{\pi^2}\right)}{n^2 e^{\left(\frac{\pi^2}{RC}\right)\tau n^2}} = \sum \frac{0.4863}{n^2 e^{0.1632(3)n^2}}$$

$$= \frac{0.4863}{(1)^2 e^{0.4896}} + \frac{0.4863}{(2)^2 e^{1.958}} + \frac{0.4863}{(3)^2 e^{4.41}} + \frac{0.4863}{(4)^2 e^{7.83}} +$$
$$= \frac{0.4863}{1.632} + \frac{0.4863}{4(7.09)} +$$
$$\frac{0.4863}{9(92.0)} + \frac{0.4863}{16(2520)} + \cdots\cdots\cdots\cdots$$
$$= 0.2980 + 0.01715 + 0.00058 + 0.00003 + \cdots$$
$$= 0.3158$$

7. $$a = \frac{1}{0.63 + \frac{2.40}{3} + \frac{3}{25.2} - 0.3158}$$
$$= \frac{1}{0.63 + 0.80 + 0.1191 - 0.3158}$$
$$= \frac{1}{1.2333} = 0.810 \text{ Btu per hour per square foot degree Fahrenheit}$$

Equation for Simple Exterior Walls

Assuming, as in the case of the simple interior wall that heat is to be supplied at a constant rate, the equation for the absorbance of an exterior wall can be expressed as:[5]

$$a = \frac{1}{R_t - \sum_{+1}^{+\infty} \frac{\left(\frac{M}{J}\right)}{e^{Qz\tau}}}$$

where

a ≡ absorbance of the wall.

$M \equiv 2RR_{s'}^2 + \frac{2R^2}{z}$

$J \equiv R^2 + RR_{s'} + R_{s'}^2 z$

$Q \equiv \frac{1}{RC}$

R ≡ the total thermal resistance of the wall exclusive of both surface resistances.

R_t ≡ total thermal resistance of the wall including both surface resistances.

R_s ≡ surface resistance of the indoor surface of the wall.

$R_{s'}$ ≡ surface resistance of the outdoor surface of the wall.

C ≡ total thermal capacitance of the wall per square foot of indoor surface.

τ ≡ time that may be allowed to elapse between the time when appreciable heat begins to enter the auditorium or room and the time when it first becomes comfortable.

z is to be found by means of the chart given in Fig. 1.

which coordinates z with $\frac{R}{R_{s'}}$

It should be noted that not only C but also R, R_s, $R_{s'}$ and R_t are per square foot of indoor surface.

The limits of the summation, +1 and $+\infty$, refer to the subscripts on M, J, and z in the next paragraph.

Procedure for Simple Exterior Walls

1. Calculate C = (density) (specific heat) (thickness of wall in feet).
2. Record R
3. Record R_s
4. Record $R_{s'}$
5. Find $R_t \equiv R_s + R + R_{s'}$.
6. Find $\frac{R}{R_{s'}}$
7. Look up the 5 values of z in Fig. 1.
8. Find Q
9. Find M_1
10. Find J_1
11. Find Qz_1
12. Find $\frac{M_1}{J_1}$
13. Repeat steps 9, 10, 11, and 12 for the subscripts 2, 3, 4, and 5.
14. Carefully check all decimal points and exponents.
15. Calculate $e^{Qz\tau}$ for each of the subscripts for $\tau = \frac{1}{4}$ hour.
16. Find the Σ.
17. Find $R_t - \Sigma$
18. Find a for $\tau = \frac{1}{4}$ hour.
19. Repeat steps 15, 16, 17, and 18 for $\tau = \frac{1}{2}$, 1, 2, 3, 4, 5, 6, 7, and 8.
20. Plot a pencil curve coördinating absorbance as ordinates against time (τ) as abscissae. This will help to eliminate some errors.

[5]Loc. Cit. Note 1 (first part).

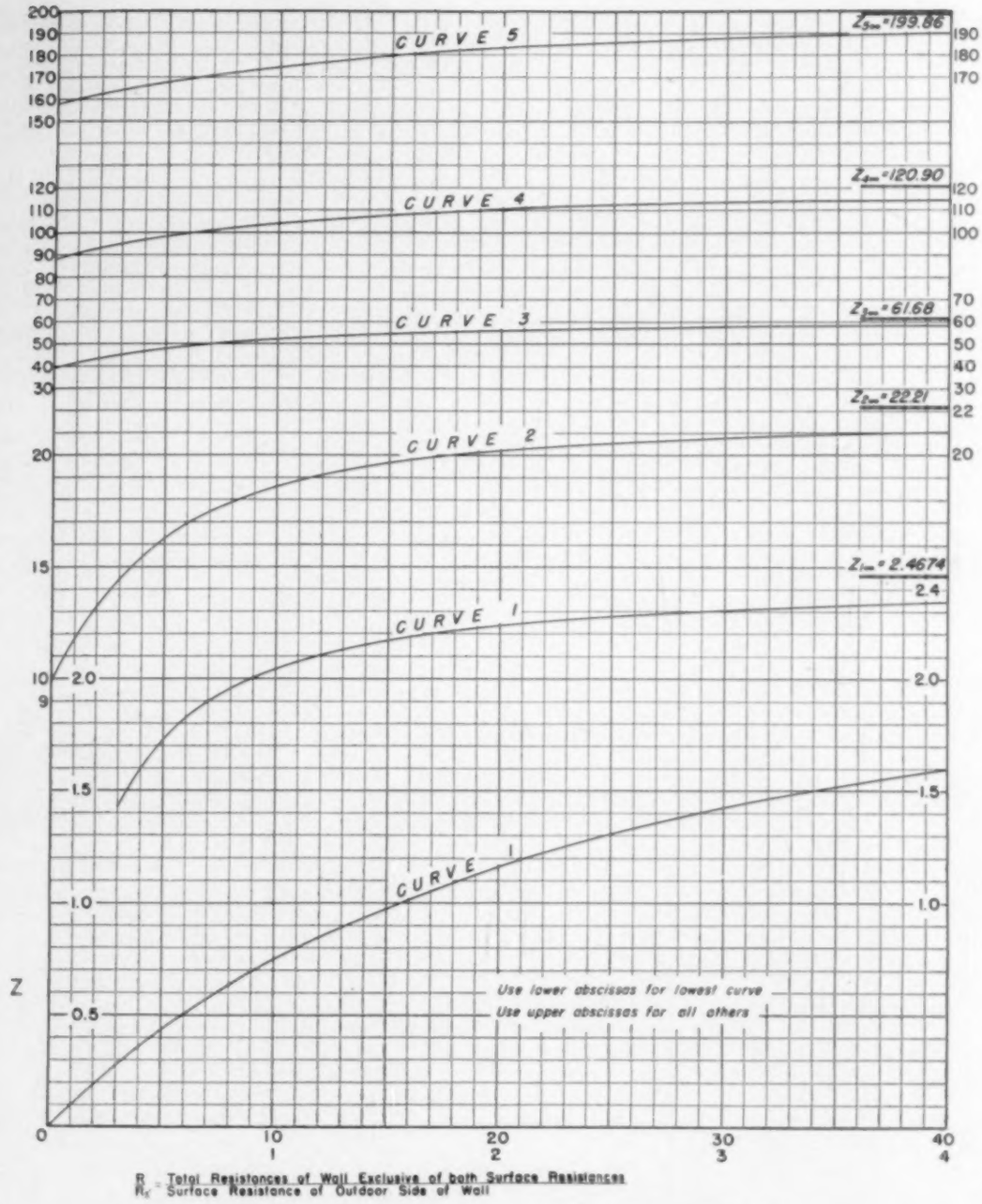

Fig. 1—Values of z for exterior wall absorbance equation

Example

Let it be required to find the absorbance of a 10-in. monolithic concrete exterior wall for various rates of heating. The steps are as follows:

1. $C = 142(0.19)\left(\frac{10}{12}\right) = 22.45$
2. $R = 0.08(10) = 0.80$
3. $R_s = 0.63$
4. $R_{s'} = 0.167$
5. $R_t = 0.63 + 0.80 + 0.167 = 1.597$
6. $\frac{R}{R_{s'}} = \frac{0.80}{0.167} = 4.79$
7. Before looking up z_1 notice that the z_1 curve is plotted in two parts. The lower part goes only from 0 to 4 so the upper part must be used to obtain z_1 for $\frac{R}{R_{s'}} = 4.79$.

$z_1 = 1.705$
$z_2 = 16.1$
$z_3 = 47.1$
$z_4 = 97.5$
$z_5 = 166.8$

8. $Q = \frac{1}{0.80(22.45)} = 0.0556$
9. $M_1 = 2(0.80)\ (0.167)^2 + \frac{2(0.80)^2}{1.705}$
$= 0.0446 + \frac{1.024}{1.705} = 0.6466$
10. $J_1 = (0.80)^2 + 0.80\ (0.167) + (0.167)^2 1.705$
$= 0.7735 + 0.0279(1.705)$
$= 0.821$
11. $Qz_1 = 0.0556\ (1.705) = 0.0950$
12. $\frac{M_1}{J_1} = \frac{0.6466}{0.821} = 0.791$
13. $M_2 = 0.0446 + \frac{1.024}{16.1} = 0.1082$
$J_2 = 0.7735 + 0.0279\ (16.1) = 1.223$
$Qz_2 = 0.0556\ (16.1) = 0.895$

$$\frac{M_3}{J_3} = \frac{0.1082}{1.223} = 0.0885$$

$$M_3 = 0.0446 + \frac{1.024}{47.1} = 0.0664$$

$$J_3 = 0.7735 + 0.0279\ (47.1) = 2.088$$

$$Qz_3 = 0.0556\ (47.1) = 2.62$$

$$\frac{M_3}{J_3} = \frac{0.0664}{2.088} = 0.0317$$

$$M_4 = 0.0446 + \frac{1.024}{97.5} = 0.0551$$

$$J_4 = 0.7735 + 0.0279\ (97.5) = 3.494$$

$$Qz_4 = 0.0556\ (97.5) = 5.425$$

$$\frac{M_4}{J_4} = \frac{0.0551}{3.494} = 0.0158$$

$$M_5 = 0.0446 + \frac{1.024}{166.8} = 0.0505$$

$$J_5 = 0.7735 + 0.0279\ (166.8) = 5.43$$

$$Qz_5 = 0.0556\ (166.8) = 9.27$$

$$\frac{M_5}{J_5} = \frac{0.0505}{5.43} = 0.0093$$

14. It is important to check all decimal points and exponents at this stage of the computations. Step 13 was given in full detail because the form used was found to result in greater speed and fewer mistakes than any other that was tried.

15. For $T = \frac{1}{4}$ hour

$e^{0.095\ (\frac{1}{4})} = e^{0.02375} = 1.0241$

$e^{0.895\ (\frac{1}{4})} = e^{0.2238} = 1.2505$

$e^{2.62\ (\frac{1}{4})} = e^{0.655} = 1.933$

$e^{5.425\ (\frac{1}{4})} = e^{1.356} = 3.88$

$e^{9.27\ (\frac{1}{4})} = e^{2.318} = 10.13$

16. $$\sum_1^5 = \frac{0.791}{1.0241} + \frac{0.0885}{1.2505} + \frac{0.0317}{1.933} + \frac{0.0158}{3.88} + \frac{0.0093}{10.13}$$

$$= 0.772 + 0.707 + 0.0164 + 0.0041 + 0.0009 = 0.8631$$

17. $1.597 - 0.863 = 0.734$

18. $$a_{\frac{1}{4}} = \frac{1}{0.734} = 1.362$$ Btu per hour per square-foot per degree Fahrenheit

19. For $T = \frac{1}{2}$ hour.

$e^{0.095\ (\frac{1}{2})} = e^{0.0475} = 1.0486$

$e^{0.895\ (\frac{1}{2})} = e^{0.4475} = 1.564$

$e^{2.62\ (\frac{1}{2})} = e^{1.31} = 3.70$

$e^{5.425\ (\frac{1}{2})} = e^{2.713} = 15.1$

$$\sum_1^5 = \frac{0.791}{1.0486} + \frac{0.0885}{1.564} + \frac{0.0317}{3.70} + \frac{0.0158}{15.1}$$

$$= 0.755 + 0.0566 + 0.0086 + 0.0010 = 0.8212$$

$1.597 - 0.821 = 0.776$

$$a_{\frac{1}{2}} = \frac{1}{0.776} = 1.290$$ Btu per hour per square foot per degree Fahrenheit.

For $T = 1$ hour

$e^{0.095\ (1)} = 1.0998$

$e^{0.895\ (1)} = 2.445$

$e^{2.62\ (1)} = 13.75$

$$\sum_1^5 = \frac{0.791}{1.0998} + \frac{0.0885}{2.445} + \frac{0.0317}{13.75}$$

$$= 0.719 + 0.0362 + 0.0023 = 0.7575$$

$1.597 - 0.758 = 0.839$

$$a_1 = \frac{1}{0.839} = 1.191$$ Btu per hour per degree Fahrenheit.

By a continuation of this process, using as many terms of the summation as are required to keep the error less than 1 per cent, is obtained:

$a_2 = 1.077$ Btu per hour per square foot per degree Fahrenheit

$a_3 = 1.001$ Btu per hour per square foot per degree Fahrenheit

$a_4 = 0.949$ Btu per hour per square foot per degree Fahrenheit

$a_5 = 0.906$ Btu per hour per square foot per degree Fahrenheit

$a_6 = 0.868$ Btu per hour per square foot per degree Fahrenheit

$a_7 = 0.839$ Btu per hour per square foot per degree Fahrenheit

$a_8 = 0.815$ Btu per hour per square foot per degree Fahrenheit

Acknowledgment

The writer wishes to express his thanks to Dean Gibb Gilchrist and to Mr. T. R. Spence for their kindness in making available to him the facilities of the Texas Engineering Experiment Station including both the equipment for making the experimental tests and the very necessary assistance in the form of National Youth Administration labor.

NAVY CALLS HOUGHTEN; JAMES IS ACTING DIRECTOR

Ferry C. Houghten is now on active duty with the United States Navy at Washington, D. C., having recently been called into service as Lieutenant-Commander, USNR. In heeding this call to serve his country, Dr. Houghten leaves the Research Laboratory activities with the best wishes of the Officers and the Committee on Research. For the past 16 years he has been Director of the ASHVE Research Laboratory at the Bureau of Mines, Pittsburgh, Pa., and has participated in a varied program of investigations sponsored by the ASHVE.

Under the stimulus of war time requirements, the Laboratory and its cooperating institutions have a big program ahead and to meet the present emergency, the appointment of John James, Technical Secretary of the Society, as Acting Director, has been announced by F. C. McIntosh, Chairman of the Committee on Research. As rapidly as possible, Society research facilities are being converted to projects of special interest to the government.

G. C. BLACKMORE, CHARTER MEMBER, DIES AT 75

George Coleman Blackmore, one of the pioneers in the heating industry, and a Charter and Life Member of the ASHVE, died at his home in Pittsburgh, Pa., on May 20, 1942, after an illness of several months. Mr. Blackmore was president of the Automatic Gas Equipment Co., Pittsburgh.

Mr. Blackmore was born on July 19, 1866 in Toronto, Ont., Canada, the son of William and Julia Stevens Blackmore, and came to New York in 1885, where he entered the heating business. With his brothers, the late Joseph J. and L. R. Blackmore, he was instrumental in organizing the American Society of Heating and Ventilating Engineers in 1894.

He organized the United States Radiator and Boiler Co., at West Newton, Pa., which in 1910 became the United States Radiator Corp., Detroit. From 1914 to 1918 he was with the National Radiator Co., Johnstown, Pa., with which was merged his own company, the Federal Radiator Co., New Castle. In 1922 the Automatic Gas Equipment Co. was organized and Mr. Blackmore became president in 1925. He was a pioneer in hot water heating and in the development of radiator heating and gas-steam radiators.

In 1931 the Council of the ASHVE made him a Life Member and until his death he was one of the few surviving Charter Members who formed the Society and helped guide it through its early days.

Mr. Blackmore married Jennie B. McIlwain in 1894 and with her made his home in Edgewood for more than 30 years. She died in 1933. He was a trustee of the Edgewood Presbyterian Church from 1926 until 1928 when he became chairman of the board. He was a thirty-second degree Mason and a member of Syria Temple.

He is survived by five children, Virginia B. Blackmore, Pittsburgh; Norman L. Blackmore, Edgewood; George W. Blackmore and Mrs. Dorothea Blackmore Lemmon, Philadelphia; and James S. Blackmore, Peru, Ind., and eight grandchildren. Funeral services were held in Pittsburgh, May 22, and burial was in Homewood Cemetery.

The Officers and Council of the American Society of Heating and Ventilating Engineers extend their sincere and heartfelt sympathy to his family for the great loss they have sustained.

St. Paul Skyline

It's Time to Go to St. Paul

THE MINNESOTA Chapter wants ASHVE members to come to St. Paul, June 15-17. R. E. Backstrom, General Chairman of the Committee on Arrangements, has sent a special message to all Society members urging them to fall in line with the conservation program and combine business with pleasure at the Society's Semi-Annual Meeting at the Hotel St. Paul, in Minnesota's Capital City.

In addition to splendid technical sessions there will be inspection trips, sightseeing and golf for men, and special trips for the ladies, including visits to the University of Minnesota Campus, the new Coffman Memorial Union and the Museum of Natural History, as well as a luncheon and card party.

A Get-Together Luncheon at noon, June 15, will be held in the Club Casino, Hotel St. Paul.

A wide variety of subjects will be discussed at the technical sessions, and of special interest will be the papers dealing with fuel conservation, air conditioning of blackout plants, and physiological reactions.

The Semi-Annual Banquet is scheduled for the evening of June 16, and the toastmaster will be Harry E. Gerrish, and the principal address of the evening will be given by Walter H. Judd, M.D.

Members have received hotel reservations cards and early requests for accommodations should be made. The rates at the St. Paul Hotel are: single $2.50 and $4.50; double $4.00 to $7.00 per day; room with twin beds $5.00 to $7.00 per day.

The Committee on Arrangements requests the cooperation of members in sending advance reservations for the three special social functions, the Monday luncheon, the dinner Monday evening and the Semi-Annual Banquet on Tuesday.

Reservations on railroad trains should be made promptly and members who desire to extend their Minnesota trip to include one or more of the national parks, will find attractive vacation rates in effect, and can receive full information about train schedules from local ticket agents. There are trips available to Yellowstone National Park, the Black Hills and Glacier National Park.

The members of the Minnesota Chapter hope to make the Semi-Annual Meeting 1942 the best in the Society's history and they say it's time now to start for St. Paul.

From	To St. Paul R. T. Rail and L. B. Pullman	R. T. Coach Fare
Atlanta, Ga.	$ 77.80	
Boston, Mass.	101.75	
Buffalo, N. Y.	67.05	
Chicago, Ill.	25.55	$15.70
Cincinnati, O.	37.45	
Cleveland, O.	53.05	
Dallas, Tex.	66.90	
Des Moines, Ia.	18.80	10.35
Detroit, Mich.	49.55	
Durham, N. C.	96.00	
Grand Rapids, Mich.	42.30	
Houston, Tex.	86.60	
Kalamazoo, Mich.	40.55	
Kansas City, Mo.	32.55	19.10
Lansing, Mich.	45.55	
Los Angeles, Cal.	135.85	
Milwaukee, Wis.	21.95	12.90
Montreal, Que.	90.70	
New Haven, Conn.	94.40	
New Orleans, La.	85.05	
New York, N. Y.	93.40	
Oklahoma City, Okla.	49.50	
Omaha, Neb.	23.40	13.95
Philadelphia, Pa.	87.50	
Pittsburgh, Pa.	62.00	
Portland, Ore.	113.85	
St. Louis, Mo.	40.10	
San Francisco, Cal.	135.85	
Seattle, Wash.	113.85	
Toronto, Ont.	65.30	
Urbana, Ill.	33.45	
Washington, D. C.	85.05	
Winnipeg, Man.	27.75	

Minnesota Chapter members take no chances in planning for golf tournament. Floyd Bell deliberates his putt, David Anderson checks the deflection, Harry Gerrish makes slide rule calculations, and Howard Betts checks the distance

Program Semi-Annual Meeting

St. Paul, Minn.—St. Paul Hotel—June 15-17, 1942

June 14

3:00 p. m. Registration and Reception of early arrivals (*Capitol Room*)

6:30 p. m. Smorgasbord Dinner (*Club Casino*)
Minnesota Movies

June 15

9:00 a. m. Registration (*Capitol Room*)

Minnesota Chapter and Ladies' Headquarters (*Sibley Room*)

9:30 a. m. Council Meeting (*Room 234*)

10:00 a. m. Nominating Committee Meeting (*Room 334*)

12:15 p. m. Get-Together Luncheon (*Club Casino*)

1:30 p. m. TECHNICAL SESSION (*Continental Room*)

Performance of a Forced Warm Air-Heating System as Affected by Changes in the Volume and Temperature of the Air Recirculated, by A. P. Kratz and S. Konzo

The Effect of Vanes in Reducing Pressure Loss in Elbows in 7-Inch Square Ventilating Duct, by M. C. Stuart, C. F. Warner and W. C. Roberts

Trend Curves for Estimating Performance of Panel Heating Systems, by B. F. Raber and F. W. Hutchinson

2:30 p. m. Ladies' Tea and Speaker—St. Paul Women's City Club

4:30 p. m. Committee on Research (*Room 234*)

7:30 p. m. Informal Get-Together (*Club Casino*)

June 16

9:30 a. m. TECHNICAL SESSION (*Continental Room*)

Overloading of Viscous Air Filters During Accelerated Tests, by Frank B. Rowley and Richard C. Jordan

Air Conditioning of Blackout Plants, by W. A. Grant

Physiologic Studies Pertaining to Naval Ventilation, by A. R. Behnke, Lt. Comdr. (MC), U.S.N.

10:00 a. m. Ladies' Sightseeing Trip—Twin Cities

Noon Ladies' Luncheon at Coffman Memorial Union, University of Minnesota, and Ladies' Sightseeing Trip—Minnetonka Lake District

12:15 p. m. Men's Luncheon at White Bear Yacht Club

1:30 p. m. Golf Tournament (Research Cup and Eichberg Memorial Cup)

6:45 p. m. Pre-Banquet Get-Together (*Club Casino*)

7:45 p. m. Dinner and Dance (*Continental Room*)
Toastmaster: Harry E. Gerrish. *Speaker:* Walter H. Judd, M.D. *Subject:* Behind the Conflict in the Pacific

Presentation of Memory Book to W. L. Fleisher

June 17

9:30 a. m. TECHNICAL SESSION (*Continental Room*)
Heat Loss Studies in Four Identical Buildings to Determine the Effect of Insulation, by D. B. Anderson

The Specific Heat of Thermal Insulating Materials, by Gordon B. Wilkes and C. O. Wood

Simultaneous Heat and Vapor Transfer Characteristics of an Insulating Material, by F. G. Hechler, E. M. McLaughlin and E. R. Queer

12:00 p. m. Ladies' Luncheon, Card Party, and Golf

12:15 p. m. Men's Luncheon at Somerset Country Club

1:30 p. m. Golf Matches and Sightseeing and Inspection Trips

Method of Choosing Location of, Financing and Conducting Meetings of the Society

Resolved: That inasmuch as the Annual and Semi-Annual Meetings of the Society come under the jurisdiction of the Council, the following rules governing the handling of such meetings be adopted by the Council and published in the JOURNAL of the Society at least twice during every year, preferably just prior to each meeting.

1—The Council will select the city in which the Annual or Semi-Annual Meeting is to be held, giving due consideration to the invitations received from Chapters or members as well as to the advisability of so distributing those meetings as to make them of the greatest advantage to the general membership, and to reduce as far as possible the expense of members attending.

2—That an appropriation be made to cover the entertainment or local expenses, incurred in connection with the meeting not exceeding $500.00, the regular meeting expense to be taken care of by the General Fund of the Society in the regular way.

3—That no registration fee or compulsory obligations of any nature be imposed on members or guests.

4—That the purchase of tickets for banquets or for any other form of entertainment that may be provided be entirely voluntary.

5—That the grouping of features and the sale of tickets for group features be discouraged.

6—That the raising of funds from manufacturers of heating apparatus be discouraged.

7—That the display of samples, or of literature, advertising the product of any manufacturer in any way, shape or form, be not permitted at the booths, registration desk, or in or about the meetings.

8—That the distribution of trade papers be entirely at the discretion of the committee in charge.

9—That the local Chapter or local members be empowered to form a General Committee with such sub-committees as may be required to handle the details of transportation, hotel accommodations, entertainment, finance, etc., and that this General Committee be requested to confer frequently with the Council, through the Secretary of the Society, and to make frequent reports on progress in connection with the various matters being handled by them.

10—That the arrangements of elaborate and costly entertainment features be discouraged.

Adopted at Council Meeting, January 29, 1926.

The Committee on Arrangements is as follows:

R. E. Backstrom, General Chairman
A. B. Algren, Vice-Chairman

Executive

N. D. Adams H. E. Gerrish
F. B. Rowley

Banquet....H. M. Betts, *Chairman*
Entertainment .....
.....E. F. Jones, *Chairman*
Finance .....
...M. S. Wunderlich, *Chairman*
Transportation .....
.....J. F. Stafford, *Chairman*
Reception .....
.....M. H. Bjerken, *Chairman*
Publicity .....
.....D. B. Anderson, *Chairman*
Ladies.....Mrs. R. E. Backstrom
Golf.....E. F. Bell, *Chairman*

BOARD OF GOVERNORS

William McNamara
G. A. Dahlstrom

NEW OFFICERS INSTALLED AT ST. LOUIS

May 5, 1942. Pres. D. J. Fagin presided at the May meeting of the St. Louis Chapter, which was held at Candlelight House in St. Louis County, with 28 members and guests in attendance. Following the reading of the minutes of the previous meeting and the usual business routine, B. L. Evans reported on the membership activities.

President Fagin reported for the legislative committee stating that the Missouri Engineers' Licensing Law had been enacted, and then announced the slate of nominations submitted. It was moved, seconded and passed that a single ballot be cast for the slate and the following were elected as officers for the 1942-43 season:

President—M. F. Carlock.
1st Vice-President—C. F. Boester.
2nd Vice-President—J. H. Carter.
Secretary—W. J. Oonk.
Treasurer—B. C. Simons.
Board of Governors—D. J. Fagin, B. L. Evans, G. B. Rodenheiser, E. H. Ronsick.

President Fagin expressed his appreciation for the cooperation received during his term as president, and the newly elected officers were inducted into office by L. Walter Moon, charter member and former president of the Chapter.

C. F. Boester, director of housing research at Purdue University, Lafayette, Ind., and member of the Society's Committee on Research, addressed the meeting on *War Housing and Post-War Housing*, stating that the main objective of the division of housing research is to invent low-cost housing. He explained that they have a 150-acre tract of land upon which the experimental houses are built. Mr. Boester gave detailed particulars involved in this work, and criticized past performance in slum clearance projects, on the ground that financing and construction are aimed at 60 to 70 years' use of the housing.

He also brought out that studies are being made toward housing of the future, in the field of radiant heating and cooling, cold cathode lighting, glass and metal wall construction, special radiation-reflecting paints, etc. A lively discussion followed, and after a rising vote of thanks to Mr. Boester, the meeting adjourned.

April 7, 1942. The April meeting was called to order by President Fagin in Candlelight House, and 40 members and guests were in attendance. Following the report on the previous meeting, the treasurer presented his report.

J. H. Carter, secretary of the Chapter, suggested that members submit ideas as to the improvement of the Society Nominating Committee procedure, so that the Chapter could then submit recommendations to the Society's Committee on Constitution and By-Laws. Mr. Boester commented on the present procedure and mentioned ways in which it could be improved.

The nominating committee of the Chapter then submitted its slate of officers for the coming year.

In place of the program that had been arranged, due to Prof. E. O. Eastwood's sudden illness in Kansas City while en route, a talking picture entitled *Railroading* was given, which was followed by adjournment.

MONTREAL HOLDS MEETING

April 20, 1942. Pres. F. A. Hamlet presided at the April meeting of the Montreal Chapter. After dinner there was a discussion on the motion made at the last meeting regarding standardization of heating design. R. F. Wormley, who is a member of the heating committee of the Wartime Price Board, suggested that the committee study the lists of the American group with regard to standardization of pipe fittings, etc. A. E. Watts mentioned that the radiation manufacturers had already agreed to reduction of available sizes.

It was moved by A. M. Peart, seconded by Mr. Wormley, that a committee consisting of S. S. Colle, F. G. Phipps and A. B. Madden, go into this more fully

E. W. Twizell, Montreal, was introduced by President Hamlet, and gave a very interesting and instructive talk on some new applications he has made of heating equipment in the hot water system recently installed in his own home. He described in detail the results he has obtained from some of these departures from standard practice, with his theories on the same, and further changes he was going to try as a result. A vote of thanks was extended Mr. Twizell and the meeting adjourned.

CINCINNATI PLANS FOR 1943 ANNUAL MEETING

April 21, 1942. Pres. W. H. Junker called the April meeting to order and the minutes of the March meeting were read and approved. T. D. Boyd then introduced R. E. Cherne who transferred from Syracuse, N. Y., to the Cincinnati Chapter.

H. A. Pillen, chairman of the meetings committee, announced that G. V. Sutfin was in charge of arrangements for the Chapter's annual meeting at Sharon Woods, with a golf tournament, a ball game and steak fry.

President Junker stated that the invitation of the Cincinnati Chapter to hold the Society's 49th Annual Meeting in Cincinnati had been accepted. It was moved by A. W. Edwards, seconded by H. W. Moore, that an election for a general chairman be held, and the motion was passed. Albert Buenger was elected general chairman, and W. C. Green, Life Member of the Society, was elected as Honorary Chairman of the Meeting.

Dr. H. C. Krueger, University of Cincinnati, then addressed the members on the subject of materials for war. After an excellent talk and discussion, the meeting adjourned.

ONTARIO CELEBRATES ANNIVERSARY

May 4, 1942. The regular annual meeting of the Ontario Chapter was held at the Royal York Hotel, Toronto, with 55 members in attendance. As this was the 21st anniversary of the Ontario Chapter, the annual meeting was combined with a celebration and an *Old Timer's Night.*

Following dinner Pres. C. Tasker called upon the various chairmen of the committees to make their annual reports, and C. D. Waldon reported on the activities of the attendance committee, while E. R. Gauley gave a report on the entertainment activities of the year.

V. J. Jenkinson, chairman of the program committee then gave a detailed report of the meetings during the past year, and suggested a short five-minute coffee talk at future meetings, which met with approval.

F. C. Moore, chairman of the auditors committee, made a report as to the financial standing of the chapter, and then called upon H. G. Hill of his committee, who explained to the Chapter exactly where their Chapter dues are applied.

D. O. Price extended a vote of thanks to all the committees for their splendid work on behalf of the Chapter. President Tasker then presented a $100.00 Dominion of Canada Victory Loan Bond to H. R. Roth in appreciation of his services rendered as secretary for the past 12 years.

President Tasker then turned the meeting over to Mr. Gauley, who acted as master of ceremonies to introduce the *old timers.* Mr. Gauley outlined the founding of the Society in 1894 and dramatically traced the growth of the Society since that time. He then called upon the Chapter's oldest member, Arthur S. Leitch, who spoke on the founding of the Ontario Chapter on May 5, 1922. Mr. Leitch outlined the difficulties experienced in forming an active chapter in Ontario, which was very interesting to the younger members.

M. F. Thomas, the first president of the Ontario Chapter, was then called upon and gave many interesting reminiscences of the first few years of the Chapter and its activities. A. J. Dickey spoke on the growth of the Society and industry and the important part it is playing in Canada today. The last *old timer* to speak was H. H. Angus, who spoke briefly on the forming of the research bureau and what it has meant to the Society. He also outlined the early days when the standards of heating and ventilating, which are every day practices today, were not established. He also brought out the self sacrifice of the early members, both in time and finances, and then told of the growth of the Society's publications.

Mr. Gauley pointed out the fast growth of the Ontario Chapter within the last few years, and called upon R. W. Bayles, chairman of the membership committee, who reported for his committee.

It was moved and seconded that the nominations for new officers be closed, and a motion to cast a single ballot to unanimously elect the ballot proposed was made, duly seconded and passed, and the following were elected for the coming year:

President—D. O. Price
Vice-President—W. C. Kelly
Secretary-Treasurer—H. R. Roth
Board of Governors—E. R. Gauley, A. S. Morgan, H. G. Hill, C. Tasker, *Ex-Officio.*

President Tasker then turned the gavel over to the incoming president, who expressed his gratitude for the opportunity of serving the Chapter as president. President Price then presented retiring President Tasker with a certificate of high esteem and appreciation of the Chapter for the excellent way he conducted the activities of the Chapter during the past year.

S. H. DOWNS VISITS NEBRASKA

March 10, 1942. The regular meeting of the Nebraska Chapter was held at the Hotel Fontenelle, Omaha, Nebr., and was called to order by Vice-Pres. C. E. Wiser.

Vice-President Wiser introduced S. H. Downs, Kalamazoo, Mich., Second Vice-President of the Society, who addressed the gathering on suggested sources of new members, and in the course of his discussion he brought out the pertinent fact that research conducted by the Society during peacetime was directly in line with the present wartime needs and could be used in war production, even though it had originally been produced for use under peace occupations

Mr. Downs then turned to the main topic of his talk which was on the subject of *Flow of Gases in Various Systems.* He pointed out that resistance to flow of gases was proportional to the temperature change, and indicated that considerable error in resistance would occur for temperature rises above 130 F, the error enlarging rapidly from 200 to 600 F.

Following an open discussion in which Mr. Downs responded to the many questions, the meeting adjourned at 9:45 p. m.

PRESIDENT EASTWOOD ADDRESSES KANSAS CITY CHAPTER

April 6, 1942. The regular meeting of the Kansas City Chapter was held at the President Hotel with 25 members and guests attending the buffet supper preceding the meeting. Pres. Gustav Nottberg opened the meeting, and the minutes of the March meeting were read and approved.

Following the treasurer's report, Vice-Pres. M. M. Rivard discussed the plans for the May meeting which was to be a question and answer type meeting.

Vice-President Rivard then introduced the guest, E. O. Eastwood, Seattle, Wash., President of the ASHVE, who presented a very interesting discussion of aerodynamic problems including a detailed discussion of wind tunnels. He pointed out that recent work is of a confidential nature so that details could not be divulged. However, he discussed the use of photographic simultaneous data readings as a decided advancement in the art of wind tunnel testing methods. He presented an interesting set of slides covering his subject and also illustrated many of the latest type airplanes used by the Army and Navy.

After a general discussion of the subject the meeting adjourned.

ATLANTA MEMBERS MEET

April 7, 1942. Following the usual dinner the meeting was called to order by Pres. L. F. Kent, at the Biltmore Hotel, Atlanta, Ga., and the minutes of the previous meeting were read. President Kent asked for committee reports and then gave some very interesting data on a pro-

posed aircraft plant to be built in the South.

The speaker of the evening, C. L. Templin, was introduced and gave a thorough discussion on the subject *Packaged Air Conditioning Units, Their Design, Application, and Difficulties.* He divided his talk into two broad classifications, comfort application and industrial application. He mentioned that the self contained units for comfort use have a wide industrial application, and are extensively used in manufacturing plants.

The highlight of Mr. Templin's presentation was the self contained unit which was set up in the meeting room and connected to an electrical supply. At the completion of Mr. Templin's talk the unit was dismantled so that the unit could be seen while functioning, and Mr. Templin then explained in detail exactly how it worked. It was an enlightening presentation and thoroughly enjoyed by everyone.

At the conclusion of the meeting President Kent told of his visit with Prof. Montgomery Knight, Aeronautical School, Georgia Tech., and a demonstration of a helicopter. The meeting adjourned at 10:15 p. m.

RADIANT HEATING DISCUSSED AT MASSACHUSETTS

April 21, 1942. Pres. J. W. Brinton called the April meeting of the Massachusetts Chapter to order at the Massachusetts Institute of Technology with 45 members and guests present, and called for Chapter nominations for the Board of Governors.

The secretary then read the list of nominations as follows: W. J. Ahearn, D. M. Archer, F. R. Ellis, G. B. Gerrish, W. A. McPherson and B. B. Howes.

F. D. B. Ingalls was introduced as a newly elected Life Member of the Society, followed by a report by the treasurer.

D. J. Edwards introduced Lt. E. J. Rodee, Quincy, Mass., who told of his experiences with four radiant heating installations. He brought out many interesting points, and following a lengthy discussion a rising vote of thanks was extended to Lieutenant Rodee, and the meeting adjourned.

CONNECTICUT MEMBERS HEAR MERRILL BLANKIN

March 10, 1942. The Connecticut Chapter of the Society held its regular meeting at the Algonquin Club, Bridgeport, Conn., and was attended by 30 members and guests.

Following dinner Merrill F. Blankin, Philadelphia, Pa., First Vice-President of the ASHVE, gave an illustrated talk on the subject of *Modern Hot Water Heating Practice.* An interesting discussion followed, and from the report from the Chapter, Mr. Blankin's talk was thoroughly enjoyed, and after a vote of appreciation to the speaker the meeting was adjourned, with the unanimous feeling of a successful meeting.

ILLINOIS ASRE AND ASHVE HOLD JOINT MEETING

April 16, 1942. The April meeting of the Illinois Chapter held at the Northwestern Technological Institute, Chicago, was a joint meeting with the Chicago section of *American Society of Refrigerating Engineers.* Dinner was served at the Institute to 215 members and guests of the two organizations.

Following dinner, Prof. B. H. Jennings, professor of mechanical engineering at Northwestern University and a member of the Chicago section of the *ASRE,* welcomed the group to the Institute and called on I. E. Brooke, president of the Illinois Chapter of the ASHVE. President Brooke introduced the officers of both chapters and adjourned the meeting to the new Institute Auditorium where Dr. L. W. Wallace delivered a timely address on the subject of research in industry.

Dr. Wallace not only brought out the many fields for research in wartime industry, but emphasized the fact that industry must also continue research for peacetime requirements with a view to post-war conditions and final settlement of world peace.

Dean O. W. Eshbach, Dean of engineering at the Institute, then described the construction of the building and its facilities for the various engineering schools housed in it.

Following these talks, members of the Engineering Staff of the Institute conducted the guests in small groups on an inspection tour of the building.

The joint meeting, as reported by M. W. Bishop, secretary of the Illinois Chapter of the Society, was most successful and the inspection tour was interesting and instructive.

DELTA STUDIES SHELTER PROBLEMS

April 14, 1942. The regular meeting of the Delta Chapter was held at Arnaud's Restaurant, New Orleans, La., with 49 members and guests present.

Through the courtesy of W. P. Oster, a technicolor sound projection film was secured entitled, *Design of a Modern Steam Generator.* A great deal of information of special interest to the members was contained in the film, which was considered a special feature of the meeting.

A report was made by G. E. May, stating that the Office of Civilian Defense in New Orleans had accepted the Delta Chapter's offer to act in the consulting capacity with regard to ventilation problems arising out of air raid shelter work. A committee of Delta members has been formed and they are studying all available data from England and other sources including information supplied by W. H. Carrier. This information is being studied with a view of recommending steps applicable to the local situation in New Orleans.

The entire Chapter membership has showed its approval of the change from evening to noon meetings for the duration, as reported by L. V. Cressy, secretary of the Delta Chapter.

CIVILIAN DEFENSE OUTLINED AT MINNESOTA

April 6, 1942. The regular monthly meeting of the Minnesota Chapter of the Society was held at the St. Paul Hotel, St. Paul, Minn., with 39 members and guests present. After a smorgasbord dinner, Pres. H. M. Betts introduced Commissioner Gus Barfuss, St. Paul, as the speaker of the evening.

The Commissioner discussed the program being inaugurated for Civilian Defense and showed sound movies of the incendiary bomb, as

well as pictures of recent bombings in London and Pearl Harbor.

The minutes of the March meeting were approved as read, followed by a discussion of the program of events planned for the Semi-Annual Meeting of the Society in June. R. E. Backstrom, General Chairman of the Committee on Arrangements for this meeting, explained the part the various committees will play in making this event a success.

M. H. Bjerken, chairman of the nominating committee for the Minnesota Chapter for the year 1942-43, gave his committee's report as follows: *President*—William McNamara; *Vice-President*—D. B. Anderson; *Secretary-Treasurer*—J. F. Stafford. Mr. Stafford then expressed his regrets that he could not accept the nomination, due to circumstances beyond his control, but it was suggested that he give the matter further consideration and advise the Chapter later. The meeting adjourned at 9:30 p. m.

FUTURE RAILROAD CARS TOPIC AT SOUTHERN CALIFORNIA

April 1, 1942. The regular meeting of the Southern California Chapter was held at Eaton's Chicken House, Los Angeles, and 50 members and guests attended. Pres. A. J. Hess called the meeting to order and called for reports from the various committee chairmen.

The nominating committee presented the following slate of officers for the 1942-1943 season: *President*—H. H. Bullock; *Vice-President*—W. O. Stewart; *Treasurer*—C. H. Weber and Myron Kennedy were both nominated; *Secretary*—Leo Hungerford.

After the business session the meeting was turned over to Mr. Bullock, chairman of the program committee, who introduced Dr. Fred C. Lindual, California Institute of Technology. Dr. Lindual spoke on the new pendulum suspended railroad cars and methods of applying air conditioning and ventilation. It was the consensus of opinion, at the conclusion of Dr. Lindual's remarks, that this will be the future method of building railroad cars and will add greatly to the speed and comfort of riding by rail. An appropriate film entitled *Railroading* followed Dr. Lindual's talk, which was an all-colored film showing the progress of railroading from the beginning up to the present era. The meeting adjourned at 10:45 p. m.

AIRCRAFT DESIGN STUDIED AT WESTERN NEW YORK

April 8, 1942. Pres. W. R. Heath presided at the April meeting of the Western New York Chapter, and the reading of the minutes opened the business session, and these were approved as read. G. E. Adema reported on the annual dance held in February and thanked his committee members for the splendid cooperation they gave to him.

H. C. Schafer, chairman of the joint meeting with the Ontario Chapter at Niagara Falls in March, submitted his report and thanked his committee for their support in making this meeting such a success. The members voiced their opinion that this type of meeting should be an annual function between the two chapters.

M. C. Beman reported for the Board of Governors and presented the list of candidates to be elected in May to serve as follows: *President*—H. C. Schafer; *1st Vice-President*—S. M. Quackenbush; *2nd Vice-President*—S. W. Strouse; *Treasurer*—B. C. Candee; *Secretary*—Herman Seelbach, Jr. In accordance with the By-Laws President Heath asked for any other nominations from the floor. It was moved and duly seconded and passed that the nominations be closed and the slate be placed before the membership for election in the usual order.

President Heath then explained to the members that owing to the fact that Prof. E. O. Eastwood was taken ill in Kansas City, it was not possible for him to be their guest. Many regrets were expressed in this announcement, and President Heath then introduced Frederic Flader, chief engineer in charge of aircraft design for a well known corporation and president of the Aero Club of Buffalo, who very kindly accepted the last minute invitation to address the Chapter in Professor Eastwood's place. Mr. Flader presented a very interesting discussion, illustrated with blackboard diagrams, showing changes and improvements effected in aircraft design as between the original model flown at Kittyhawk and the present day aircraft. He pointed out that, while there were no essential differences in principle as between the old and the new aircraft, the great improvements had come through streamlining, power plant design, variable pitched propellers, etc., which had tended to decrease the drag and increase the engine performance. He also vividly illustrated the tremendous increase in fire power of military machines as between the First World War and the present equipment. His talk was most instructive and well received by the entire membership.

A record turn-out of members and invited guests expressed themselves as having been well satisfied with the program offered, although deep regret was expressed over the unfortunate illness of Professor Eastwood.

GOLDEN GATE HEARS PRESIDENT EASTWOOD

April 4, 1942. In the absence of Pres. C. E. Bentley, Vice-Pres. Jared Hill took the chair and called the April meeting to order in the Bellevue Hotel, San Francisco, Calif., with 26 members and guests in attendance.

The acting chairman called on N. H. Peterson and R. A. Folsom to introduce the guests, followed by the minutes of the previous meeting which were approved as read.

After the report by the treasurer, G. M. Simonson, chairman of the nominating committee, which was appointed at the February meeting, submitted the report from the nominating committee recommending the following officers: *President*—J. F. Kooistra; *Vice-President*—R. A. Folsom; *Secretary*—C. L. Peterson; *Treasurer*—R. B. Holland; *Board of Governors:* James Gayner, Mason Emanuels and Oliver Johnson.

The chairman called for nominations from the floor, and there being none, instructed the secretary to mail the ballots to the membership, and appointed Messrs. P. R. Babcock, E. H. Goins and F. W. Hook as tellers to record the ballot at the annual Chapter meeting.

Mr. Hill as chairman of the program committee presented a most

interesting technicolor film showing the entire history of the discovery and production of aluminum. It was also educational and was enjoyed as evidenced by the discussion. The meeting adjourned at 1:45 p. m.

March 4, 1942. The March meeting was held at the Bellevue Hotel, San Francisco, and the minutes of the January meeting were dispensed with. A nominating committee was appointed by President Bentley consisting of G. M. Simonson, *chairman,* R. A. Hudson and N. H. Peterson.

E. W. Simons reported on the committee appointed to confer with the San Francisco Civilian Defense Council offering assistance of the Chapter. Conferences were held with Mr. Litchfield by Messrs. Simons and Kooistra.

President Bentley then reported on the Annual Meeting of the Society in Philadelphia and emphasized the necessity of an active chapter with fully paid-up membership.

Mr. Hill mentioned that he was endeavoring to get Major Osborne, Assistant Coordinator of the 9th Area Civilian Defense, to address the Golden Gate Chapter on the relation of the heating and ventilating engineers to Civilian Defense problems.

Prof. E. O. Eastwood, President of the Society, was the guest of the evening and was introduced by Mr. Hill. Professor Eastwood's subject was *Heating and Ventilation of Modern Aircraft,* with side references to Society activities. He illustrated his talk with a number of slides, showing flight of birds, insects and aircraft and the use of the wind tunnel at the University of Washington. It was an extremely interesting talk and was thoroughly enjoyed by all present. The meeting adjourned at 2:00 p. m.

SOLAR HEAT DISCUSSED AT WASHINGTON, D. C.

April 8, 1942. The regular meeting of the Washington, D. C., Chapter was held at the Dodge Hotel, with 25 members and guests in attendance. Following the regular order of business and the reports from the various committee chairmen, the meeting adjourned to the Garden House where Squadron Leader R. L. Winterburn of the R.A.F. gave an extemporaneous talk on the effect of war on the engineering profession in England.

L. H. Holder then delivered a very instructive talk on *Reduction of Solar Heat Load by Water Spray,* and presented numerous charts and pictures illustrating his statement that a properly designed roof spray system, adequately controlled, would materially improve temperature conditions of the floor immediately under the roof as well as retard the disintegration of the roof surface itself.

J. H. Clark, acting chairman of the nominating committee, announced that the names of the following had been placed in nomination for officers for the coming year: *President*—R. S. Dill; *Vice-President*—F. M. Thuney; *Secretary*—J. W. Markert; *Treasurer*—F. B. Sale; *Board of Governors*—F. A. Leser, S. L. Gregg, W. H. Littleford.

The Chapter approved a motion to have the chairman of the meetings committee present for the approval of the Chapter at the next meeting, an instruction sheet to acquaint guest speakers with the ethics of the Society and of the Washington, D. C., Chapter.

President Leser presented a beautifully embossed holder for the minutes of the Chapter meetings, and the secretary was intructed to send a letter of appreciation to Lt. Commdr. T. H. Urdahl, who donated the book. The secretary also thanked President Leser for his part in having the book embossed with the Chapter's name. The meeting was declared adjourned at 10:00 p. m., according to F. M. Thuney.

Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 32 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by June 15, 1942, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ALVAREZ, JOAQUIN, JR., Engr., Pittsburgh Lectrodryer Corp., Pittsburgh, Pa.	G. L. Simpson	H. E. Zankey (*Non-member*)
ANDREWS, WILLIAM R., Asst. Mgr., Ross Engineering of Canada, Ltd., Montreal, Que., Canada.	W. E. Moore (*ASME*)	H. G. Heath (*Non-member*)
BAKER, THOMAS A., Vice-Pres., Baker Specialty & Supply Co., Logansport, Ind.	R. R. Noyes	A. F. Hamlet
BAZZONI, JOHN P., Mech. Engr., G. P. Rice Co., Biloxi, Miss.	C. W. Johnson	Leo Garneau
BLANCHARD, NORRIS M., Western Sales Mgr., L. J. Mueller Furnace Co., Milwaukee, Wis.	H. A. Thrush	K. T. Davis
BOWEN, LEROY F., Mgr. (Htg. Div.) Gate City Petroleum Co., Kansas City, Mo.	W. R. Stockwell	A. G. Dixon
	A. R. Salzer, Jr.	J. J. Friedler, Jr.
	G. C. Kerr	N. J. Helwick
	D. D. Williams	C. E. Wiser
	H. W. Stanton	B. G. Petersen
	J. B. Fehlig, Jr.	F. F. Dodds
	E. K. Campbell	J. M. Arthur

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BRYNER, JOHN J., Chief Engr., Roosevelt Hotel, New Orleans, La.	W. H. Dudley, Jr.	G. C. Kerr
CLARKE, JOHN H., Marine Engr., U. S. Maritime Commission, Washington, D. C. (*Advancement*)	G. E. May	John Devlin
CUTTING, RICHARD H., Archt.-Engr., Garfield, Harris, Robinson & Schafer, Cleveland, O.	F. A. Leser	R. S. Dill
FAGAN, LAWRENCE E., Sales Engr., Chatham Malleable & Steel Products Ltd., Toronto, Ont., Canada.	F. M. Thuney	G. R. Walz
FINLAY, ALVIN E., Htg. Engr., Portland Gas & Coke Co., Portland, Ore.	L. T. Avery	G. P. Nachman
HARRIS, WARREN S., Research Engr., University of Illinois, Urbana, Ill.	J. P. Jones	P. D. Gayman
HOLDEN, ROBERT G., Mech. Engr., Wilbur Watson and Associates, Ravenna, O.	G. C. Kemp	H. R. Roth
HUFF, JAMES M., Mgr. (Mech. Dept.), Robert & Co., Inc., Corpus Christi, Tex. (*Reinstatement*)	V. J. Jenkinson	F. C. Moore
LOCHMAN, EDWARD W., Owner, Edward W. Lochman Co., Kansas City, Mo.	B. W. Farnes	T. E. Taylor
MACFARLAN, NORRIS S., Sales Engr., Philadelphia Gas Works Co., Philadelphia, Pa.	E. E. Carroll	J. D. Kroeker
	M. K. Fahnestock	S. Konzo
MCFARLAN, ALDEN I., Sales Engr., Kerby Saunders, Inc., New York, N. Y.	A. P. Kratz	W. H. Severns
MORGAN, EDWIN H., JR., Draftsman, Chemical Construction Corp., New York, N. Y.	C. F. Eveleth	R. A. Wilson
PATTERSON, FRANK H., Sales & Service Engr., Hoffman Specialty Co., Indianapolis, Ind. (*Reinstatement*)	L. H. Pogalies	N. H. Hall
PEARSON, FRED L., Asst. Chief Mech. Engr., Giffels & Vallet, Inc., Detroit, Mich. (*Reinstatement*)	L. S. Gilbert	L. C. McClanahan
ROBERTS, EDWARD F., JR., Partner, Edward F. Roberts Co., Philadelphia, Pa. (*Reinstatement*)	T. H. Anspacher	J. A. Bishop
SCHULEIN, LARS E., Sales Engr., Economy Electric Manufacturing Co., Chicago, Ill.	P. C. Leffel	M. M. Rivard
SHELBY, ALEXANDER W., Engr. & Partner, Shelby-Skipwith, Inc., Memphis, Tenn.	L. A. Stephenson	W. L. Cassell
SHELDON, ROBERT W., Sales Engr., The Lake Shore Gas Co., Ashtabula, O.	Mortimer Traugott	A. B. Eckert (*Non-member*)
SOULE, JOHN P., Mech. Engr., Federal Shipbuilding & Dry Dock Co., Newark, N. J.	A. G. Dome	F. W. Thernholz (*Non-member*)
TRIEGEL, ERICH V., Mech. Engr., Francisco & Jacobus, New York, N. Y.	H. J. Ryan	R. A. Wasson
TUTT, RICHARD D., Air Conditioning Engr., General Motors Corp. (Allison Div.), Indianapolis, Ind.	R. R. Ferguson	D. A. MacWatt
WAGNER, WILBUR J. O., Owner, Wagner Temperature Store, Jefferson City, Mo.	A. H. Bermel	F. E. W. Beebe
WANGSGAARD, DEE, Prop., Wangsgaard Coal & Stoker Co., Logan, Utah.	L. B. Ray	James Tiernan
WEISS, EDWARD J., Asst. Industrial Air Cond. Engr., General Electric Co., Schenectady, N. Y.	M. F. Blankin	Ferdinand Jehle
WIEDENMANN, WILLIAM A., Owner, W. C. Wiedenmann & Son, Kansas City, Mo.	M. C. Gillett	M. B. Shea
WILLIAMSON, CHESTER C., Operator, H. J. Heinz Co., Pittsburgh, Pa.	T. S. Lovell (*ASME*)	C. E. Stillson (*ASCE*)
	Herman Cohen (*ASME*)	J. J. Hayes
	M. F. Blankin	C. F. Dietz
	H. H. Erickson	A. E. Kriebel
	V. L. Sherman	E. H. Wendt
	J. W. Mabbs (*Non-member*)	M. K. Arenberg
	J. R. Hertzler	S. F. Nicoll
	L. J. DuBois	V. T. Kartorie
	J. A. Schurman	W. R. Beach
	C. A. McKeeman	R. W. Geltz
	L. C. Soule	W. E. Heibel
	W. H. Carrier	W. L. Fleisher
	O. O. Oaks	Joe Wheeler, Jr.
	Henry Gitterman	M. F. Rather
	G. B. Supple	R. A. Stuart (*Non-member*)
	E. S. Hildreth	S. E. Fenstermaker
	F. J. Dean, Jr.	Carl Clegg
	P. C. Leffel	K. M. Stevens
	Delbert Berntson (*Non-member*)	B. B. Blair (*Non-member*)
	W. R. Tolman (*Non-member*)	E. M. Stock (*ASCE*)
	R. B. Taylor	J. S. Leibson (*Non-member*)
	C. E. Hunziker	C. A. Linden (*AIEE*)
	W. L. Cassell	J. B. Fehlig, Jr.
	N. W. Downes	P. C. Leffel
	E. C. Smyers	W. H. Reed, III
	T. F. Rockwell	F. C. McIntosh

Candidates Elected

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

BRODIE, AARON H., Pres., J. Brodie & Son, Inc., Detroit, Mich.

CHAPPELL, HENRY D., Mech. & Elec. Engr., Burroughs Adding Machine Co., Detroit, Mich. (*Reinstatement*)

DROPKIN, DAVID, Instructor, College of Engrg., Cornell University, Ithaca, N. Y.

DUKEHART, MORTON MCI., Proprietor, Morton McI. Dukehart & Co., Baltimore, Md.

ELLINGSON, E. T. PALMER, Consulting Engr., Oklahoma Architects, Oklahoma City, Okla.

GOULD, HENRY E., Partner, Natkin & Co., Kansas City, Mo. (*Reinstatement*)

HAMILTON, M. S., Dist. Mgr., Minneapolis-Honeywell Regulator Co., Pittsburgh, Pa.

HASTINGS, ADDISON B., Asst. Secy.-Sales Engr., Burnham-Boiler Corp., Irvington, N. Y.

HEDEEN, LAUREL E., Associate Engr., M. L. Todd & Associates, Waterloo, Ia. (*Advancement*)

JENNINGS, BURGESS H., Prof. Mech. Engrg., Northwestern Technological Institute, Evanston, Ill.

MADDEN, ALFRED B., Mgr. (Htg. Div.), Crane Ltd., Montreal, Que., Canada.

MCCAUL, LYNN K., Sales Engr., The Coon-Devisser Co., Detroit, Mich.

MUSSER, JOHN M., Engr. (Htg. & Vtg.) U. S. Engineers, Rome, N. Y.

NORTH, CLARENCE P., Chief Engr., Campbell Heating Co., Des Moines, Ia.

OREAR, ANDREW G., Trade-wind Motorfans, Inc., Los Angeles, Calif. (*Reinstatement*)

REISBERG, LESTER K., Vice-Pres. Goodin Co., Minneapolis, Minn. (*Advancement*)

SHEPPARD, WALLACE K., Heating Engr., The Peoples Natural Gas Co., Pittsburgh, Pa.

TAYLOR, THOMAS E., Consulting Engr., Portland, Ore. (*Advancement*)

THOMAS, ERNEST R., Mech. Engr., Black & Veatch, Consulting Engrs., Ft. Smith, Ark.

TRICKLER, EARL E., Dist. Sales Mgr., New York Blower Co., Chicago, Ill.

TYDINGS, WILLIAM F., Partner, Tydings Engineering Co., Detroit, Mich.

WEBSTER, WILLIAM H. JR., Sales, Allied Heating Products Co., Norfolk, Va. (*Reinstatement*)

ASSOCIATES

ALLEN, WILLIAM A., Vice-Pres., Sprague & Sprague, Inc., Pittsburgh, Pa. (*Reinstatement*)

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